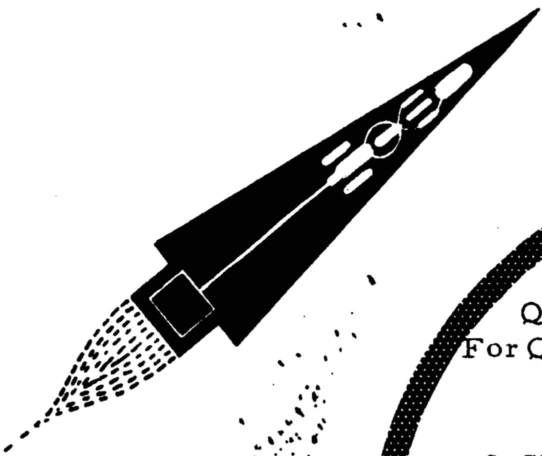
SPACE POWER OPERATION



QUARTERLY REPORT NO. 2
For Quarter Ending November 8, 1961

TWO-STAGE POTASSIUM TEST TURBINE

Under Contract NAS 5-1143

for

THE NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FLIGHT PROPULSION LABORATORY DEPARTMENT

GENERAL ELECTRIC

SPACE POWER OPERATION

SECOND QUARTER PROJECT STATUS REPORT

NOVEMBER 8, 1961

TWO-STAGE POTASSIUM TEST TURBINE

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

CONTRACT NUMBER NAS 5-1143

FLIGHT PROPULSION LABORATORY DEPARTMENT

GENERAL ELECTRIC COMPANY

CINCINNATI 15, OHIO

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I. SUMMARY

The Flight Propulsion Laboratory Department of the General Electric Company has been under contract to the National Aeronautics and Space Administration since May 8, 1961, for the design and fabrication of a two-stage test turbine suitable for operation in saturated potassium vapor at 1600°F. The test turbine consists of stages three and four of a five-stage 500 KW turbine and is to have a design flow capacity of 2.8 pounds per second. The present one-year contract covers design and fabrication of the turbine.

The main objectives of this program are to study the effects of vapor wetness on performance, to demonstrate interstage condensate extraction, to study blade erosion with different blade materials, to study the phenomena of supersaturation and droplet formation, to establish correct potassium vapor properties as an improvement over G.E.'s calculated Mollier diagrams and finally, to establish accurate fluid flow design methods for potassium turbines operating in the wet vapor region. The test turbine runs on oil lubricated bearings. The test program anticipates 20 hours of performance testing, 1,000 hours of endurance testing under design conditions at 19,200 rpm and 1,000 hours endurance testing under aggravated erosive conditions at 24,000 rpm.

The present report covers progress during a three-month period, ending November 8, 1961. The main events of this reporting period are:

Fluid Design

The final fluid dynamic design of the nozzle partitions and buckets of both stages has been completed.

Mechanical Design

The overall layout drawing of the potassium test turbine has been changed since the last quarterly report to reflect changes required by the detailed design of many of the component parts. The design of both the inlet and exhaust sections has been modified to improve the installation of the test turbine into the facility. Design improvements have been made on virtually all rotor and stator component parts to obtain greater strength, ease of assembly, lower cost, and improved maintainability. The emphasis during this reporting period has been directed toward the completion of the detailed manufacturing drawings of all turbine components. However, special emphasis has been placed upon completion of the detailed drawings of the fluid dynamic components because of their greater complexity and the fact that they are made of materials requiring the longest procurement cycles.

Buckets have been designed for each stage which may be replaced individually. They have resulted in increased dovetail stresses and decreased bucket natural frequencies. In a comparative design study a horizontally-split casing design has been selected over a circumferentially-split design because the former presents a lesser problem in axial and concentricity accumulative tolerances, has fewer parts, less cost, a more direct path for instrumentation leads and requires 36 per cent less flange sealing length than the latter. 316 stainless steel (the facility material) has been selected for the casing over L-605 because this choice reduces the thermal stresses at the flanges where the test turbine and facility join.

A considerable number of detailed manufacturing drawings have been completed with detailed notes as to fabrication methods and sequences, heat treatment, and special requirements because of the alkali-metal atmosphere. Stress analyses in progress or completed include: the nozzle diaphragm assemblies, rotor blades, and tie bolt. A tentative design has been completed of a glove box to be placed around the test turbine to permit inspection and replacement of buckets and possibly other parts of the test turbine without oxygen contamination of the test turbine or the facility.

Tests are being conducted to determine the proper machining and heat treatment cycle required for broaching the dovetail slots in the turbine discs. Tests simulating the brazing of L-605 nozzle partitions into 316 stainless steel shroud bands have been completed and the results are being evaluated.

Procurement has been initiated on all parts for which detailed fabrication drawings have been completed.

Instrumentation

The required performance and steam pretesting of pressure measuring devices has been completed. Tests have been conducted to establish the flow of argon per tap through the efflux pressure measuring devices both for normal operation and for instrumentation purging.

Test Facility

A design of the building addition needed to house the test facility has been initiated. Five boiler designs are being evaluated for acceptance and a purchase order will be placed in a few weeks. The boiler will have a 316 stainless steel tube bundle and will produce a quality of 98 to 99 per cent without a superheater. Two condenser configurations are being evaluated and a purchase order will soon be placed for this component. At present, an attempt is being made to obtain a surplus eddy current dynamometer rated at 3000 horsepower up to 24,000 rpm.

An alternate solution is an open-cycle system consisting of a turbo-supercharger with variable inlet guide vanes.

Materials Support

The materials selection has been reviewed, and specifications have been prepared for the procurement of the turbine and test facility materials. Several experimental investigations were initiated to examine fabrication procedures and to obtain more extensive documentation of some of the materials which will be used in the turbine. The TZM and F-48 refractory alloy turbine blade materials were obtained, and their evaluation is in progress.

Schedule and Forecast

In the most recent quarter certain unanticipated design refinements were deemed necessary to assure the highest possible degree of success in the test program. For example, the buckets of both stages were redesigned to permit individual replacement. Both circumferentially and horizontally-split casings were investigated to obtain the best solution to the problem of bringing the instrumentation leads out of the casing and to the problem of reducing flange leakage length. To enhance the reliability of the test vehicle a symmetric scroll was designed to replace an offset configuration. Studies have been made of a glove box around the test turbine to permit inspection and part replacement without contaminating the turbine or facility with air. These and other important design studies have delayed slightly the completion of the test turbine mechanical design in layout and detail drafting. In turn, the initiation of the procurement and fabrication cycles has been delayed until the month of December. The revised schedule shown in Table VII indicates that by January 8 the test turbine mechanical design and detail drafting will be complete. Since long fabrication lead times were provided for in the schedule the completion date of the assembly of the test turbine is shown unchanged in the schedule.

At the end of the third quarter, February 8, 1962, the mechanical design, layout and detailing of the test turbine will be complete and orders will have been placed for forgings, blades and major items of test equipment. In addition the power brake will have been selected and most of the test facility design and drafting will be complete. The test plan also will have been formulated. Manufacture of test turbine and facility will be well under way.

II. FLUID DYNAMIC DESIGN

During the reporting period certain detail changes have been made in the blading to facilitate the mechanical design. However, the velocity vector diagram data presented in the previous quarterly report remains unchanged.

A. NOZZLE PARTITIONS

In the last reporting period it was decided that a twisted nozzle partition had several fluid-dynamic and mechanical design advantages over the untwisted ones reported in the first quarterly progress report. Among these advantages are reduced unguided turning at the discharge and ease of manufacture. The final nozzle-partition profiles for stages one and two are shown in Figures 1 and 2, respectively. In both stages the nozzle-partition profiles at all radial locations are the same, but are stacked about the pressure (concave) surface trailing edge. The nozzle partition profiles have thick blunt leading edges. This shape is insensitive to performance changes over wide ranges of angle of attack and resists moisture droplet erosion.

Due to space limitations, the nozzle partitions in stage number two had to be reduced by increasing the number from 34 to 38 (solidity remains unchanged). The number of nozzle partitions in stage number one, however, remained the same, namely: 46.

B. BUCKETS

Shown in Figures 3 and 4 are comparisons between the bucket blade drawings presented in the first quarterly report and the final configurations selected for stages one and two, respectively. Due to the fact that the blade trailing edges of the preliminary configurations for both stages overhung the bucket platform, the buckets could not be inserted into and removed from the turbine discs individually. However, because of the corrosive, erosive environment in which these buckets must operate, frequent inspection is mandatory. Therefore, a redesign of the bucket blading to remove the overhang was required. The result was the final configurations shown in Figures 3 and 4 In the final configurations shown, a small positive clearance is provided which permits the individual insertion and removal of the buckets in each stage.

The obtaining of the final configurations of the buckets was complicated by the high hub turning angles (129.6 and 125.2 degrees, respectively, for stages one and two) required by the velocity-vector diagrams and marginal disc dovetail stresses caused by heavy bucket blade sections.

The overhang is a direct result of the high turning angles and high solidity used. Intermediate designs indicated that reducing the bucket hub section thickness resulted in reduced bucket weight and dovetail stress but increased trailing edge overhang. Table I summarizes pertinent data concerning the preliminary and final configurations.

Shown in Figures 5 and 6 are the final hub, pitch, and tip bucket blade sections for stages one and two respectively. These blade sections were obtained indirectly through the use of the velocity-vector diagram data reported in the first quarterly report in the contractor's cascade analysis digital computer program. The latter program calculates the potential flow around an arbitrary cascade of airfoils specified as input. The result is the surface-velocity distribution around the airfoil. Profiles having unfavorable surface velocities distributions are modified until a satisfactory distribution results.

Shown in Figure 7 is a surface velocity distribution which is typical of those obtained for the final blade shapes. Specificially, Figure 7 is the surface velocity distribution for the hub section of the second-stage bucket. Aside from slender cusps near the leading and trailing edges (which are merely due to discontinuities in the radii of curvature of the specified airfoil shape) the velocity distribution of Figure 7 is certainly favorable. On the suction (convex) surface, a steady acceleration takes place until well downstream of the blade throat, where a deceleration is always expected.

On the pressure (concave) surface, a steady acceleration from the 40 per cent chord position to the trailing edge stagnation region assures good performance of this blade section. The velocity distribution shown in Figure 7 for the final configuration compares favorably with that of the preliminary configuration shown in Figure 8.

Shown in Figure 8 is the surface velocity distribution for the hub section of the second stage bucket reported in the first quarterly report for the preliminary configuration.

Shown in Table II are physical dimensions of the final test turbine buckets.

III. MECHANICAL DESIGN

A. GENERAL CONFIGURATION

Shown in Figure 9 is the overall layout of the potassium test turbine. While it shows the same basic configuration and overall dimensions as that presented in the previous Quarterly Report, the present design incorporates a number of detail changes which have been made as the turbine design has progressed. For example, it will be noted that the design of both the exhaust and inlet sections have been modified to improve the overall turbine - facility installation, and to minimize thermal stresses.

Shown in Figure 10 is an enlarged cross-sectional drawing of the rotor and stator assembly, which, except for minor dimensional modifications, represents the final turbine configuration. Design improvements have been made on virtually all rotor and stator component parts to obtain greater strength, ease of assembly, lower cost, and improved maintenance characteristics.

The primary design effort continues to be directed toward completing detailed manufacturing drawings of all turbine components for manufacturing release, with special emphasis upon the components of greatest complexity and those requiring the longest procurement cycle.

B. DETAILED DESIGN

1. Blading Design

In Figures 3 and 4 are shown the front and top view of the buckets for stages 1 and 2, respectively, including the airfoil, platform, and dovetail. Both preliminary and final configurations are shown. As mentioned previously, the bucket blade shapes for the final buckets are shown in Figures 5 and 6. One goal in the mechanical design of these blades has been that of the individual replacement of rotor buckets, which allows for the maximum ease of maintenance and inspection. Additional mechanical requirements have been geometric simplicity to facilitate machining of the buckets from solid bar stock, and high strength, necessary for the long operating life of the turbine. In the preliminary blading designs, the fluid design required tangential overlapping of adjacent airfoils, with a result that under no circumstances could the turbine buckets be replaced individually. In addition, it was required that either the bucket platform of the second stage have a bent platform which would allow it to turn and thereby extend under the airfoil trailing edge, or, alternatively, that the airfoil be allowed to overhang the platform at its trailing edge. The former approach involved considerable manufacturing problems, and the latter required a small radius undercut of the root of the airfoil trailing edge to blend into the platform. This was undesirable from the standpoint of performance, and produced a source of stress concentration.

In the present blading shown in Figures 3 and 4, both manufacturing simplicity and individual replaceability of the blades has been obtained with no loss in fluid dynamic performance. In the final bucket design, however, the weights of the buckets of each stage have increased over those weights of the preliminary design, resulting in higher dovetail stress and in lower natural frequencies of vibration for the respective buckets.

A single dovetail form has been selected for both stages. It is the dovetail of a turbojet engine presently in production at General Electric. Glassines showing a 50 times magnification of the bucket dovetail form and wheel slot have been obtained and are being used in the dovetail stress analysis and to determine the bucket center of gravity location.

As pointed out previously, the nozzle partitions of stages 1 and 2 have been modified to a configuration having the constant cross-sections spanwise shown in Figures 1 and 2, and a small amount of twist. The nozzle diaphragm drawings have been modified as required. The number of nozzle vanes for Stages 1 and 2 are 38 and 46, respectively.

2. Casing Design

An extensive design review of the turbine casing configuration has been made in the light of the most recent knowledge of stress levels, manufacturing techniques, instrumentation requirements, and maintenance problems. The result has been a detailed and critical comparison between the horizontally and circumferentially flanged casing configurations shown in Figures 10 and 11, respectively.

Because of the requirement that the pressure sensor leads be attached to the casing and nozzle partitions by brazed or welded joints only, and because cutting and re-welding of pressure sensor leads each time the casing is opened is not desirable, a requirement has been established to design for the assembly and disassembly of the turbine without disturbing the instrumentation. In addition, it is highly desirable to instrument the test turbine before installation in the facility to avoid such techniques as threading leads through the casing and brazing them at the site of final assembly. Shown in Figure 11 is the circumferentially split casing in which the above requirements are met. This is accomplished by extending the outer shroud bands (which carry the pressure sensor leads) of the nozzle-diaphragm through the casing by means of "sandwiched" flanges. All of the pressure leads can be brazed to the nozzle diaphragm assembly at the same time that the partitions are brazed to the shroud bands. This permits the instrumentation of the turbine during fabrication rather than during assembly. With regard to the horizontally-split casing of Figure 10, the pressure sensor leads extend directly through the casing wall, and are also inserted at a bench when the horizontally split diaphragms are mounted into the split casing. In this case also, the instrumentation of the turbine is accomplished during fabrication of the parts rather than during final assembly.

The circumferentially flanged casing would normally be more desirable from the standpoint of eliminating the leakage problem associated with the horizontally flanged casing at the inevitable junction of the horizontal and circumferential flanges. However, a requirement which has been imposed in the light of previous turbine experience is that it must be possible to disassemble and reassemble the turbine without disassembling the already balanced rotor. In the circumferentially-flanged design, the second nozzle diaphragm must be split along the horizontal centerline because it is located between the two turbine wheels. Therefore, the sandwiched flange of the second stage nozzle diaphragm becomes in effect a horizontally-split casing of very short length. This short horizontal split is as difficult to seal as a full horizontally-split casing.

The re-evaluation of the casing designs reveals that lower axial and concentricity stack-up errors, lower cost, fewer parts, a more direct path for instrumentation lines, and 36 per cent less flange sealing length are obtained through the use of the horizontally-split casing shown in Figure 10. For these reasons it has been decided to adopt the horizontally split casing since the corner sealing problem occurs in both designs. This approach was selected despite greater problems of maintaining roundness with the horizontally split casing. It will be alleviated by making the casing wall relatively thick, as shown in Figure 10.

As a result of the investigation of casing stresses, it has been concluded that the casing can be made of 316 stainless steel, which is the facility material, rather than the L-605 previously considered. One of the primary reasons for the selection of the 316 stainless steel for this component is to eliminate the possibility of large flange stresses at the hot turbine inlet resulting from the differential thermal expansion of dissimilar materials. Any such stresses produced in this critical region of the turbine casing could easily offset the advantage of the greater strength of L-605.

The turbine casing is subjected to combined hoop, bending, and thermal stresses. The precise levels of these stresses are difficult to evaluate because of the complexity of the geometric shapes of the turbine and attached ducting, and because the turbine is part of the closed system subject to thermal expansion. Various techniques are being employed to minimize the casing stresses, with primary consideration being given to the inlet section where higher values of temperature and internal pressure are encountered. In addition to minimizing the combined stresses at the casing inlet by increased wall thickness, the casing bending stresses are being minimized by mounting the turbine on a dolly, which is free to move with thermal expansion of the duct between the boiler and the turbine inlet. The condenser is carried along with the turbine on the dolly to minimize bending of the exhaust duct. In addition, the main turbine mount is located near the exhaust scroll where it can absorb any moment caused by exhaust duct misalignment, thus minimizing the moments which reach the turbine casing at the inlet flange. The inlet casing flange is supported directly from the base plate to prevent the weight of the turbine inlet duct from acting on the casing. Insulation will be used around the turbine to minimize thermal stress in the casing wall and to minimize internal surface condensation.

Because of the requirement of holding small clearances between rotating and static parts, the casing must be designed to creep limits, rather than to stress rupture limits which are pertinent to the facility ducting design. An investigation of type 316 stainless steel creep strength reveals that for 1 per cent creep in 1000 hours at 1600°F, the allowable stress is approximately 1600 psi based on minimum properties. The design goal for the casing is to maintain the combined stress at values less than 1000 psi at the inlet.

Because of moderate internal pressures, and the use of the heavy casing walls, the casing hoop stress is less than 300 psi. The anticipated bending stress is also less than 300 psi. Thus, the combined stress will be less than the allowable stress of 1000 psi.

3. Exhaust Scroll

The previous exhaust scroll design (Figure 13a) was a non-symmetrical involute design with the single exhaust duct center line located about nine inches away from the turbine center line. With the condenser located on swinging supports directly below the turbine and connected by a twelve-inch diameter rigid duct (see Figure 12), any thermal expansion of the scroll produced a bending load on the duct and scroll, and any change in condenser weight produced an unbalanced force along the duct, resulting in a torsional moment on the scroll because of the nine-inch offset. A redesign of the turbine exhaust scroll was, therefore, made to reduce or eliminate the stresses and deformations in the turbine resulting from the thermal expansion of the closed loop ducting under operating conditions.

The redesigned scroll (Figure 13b) comprises two symmetrical involute sections exhausting into a duct whose centerline is in the same plane as that of the turbine, thus eliminating lateral forces on the duct due to thermal expansion, and scroll torsional moments due to forces acting along the exhaust duct. The main turbine mount connecting the turbine to its support platform is located near the scroll aft flange as shown in Figure 12. This prevents axial thermal expansion of the turbine from imposing lateral forces on the exhaust duct, and insures that any loads transmitted to the turbine through the exhaust duct will be largely absorbed by the turbine support, and not transmitted through the casing to some remote supporting structure.

4. Detailed Drawings

In addition to the layout drawings of Figures 9 and 10, detailed manufacturing drawings have been completed for most of the turbine components. Emphasis has been placed upon completing the complex fluid dynamic assemblies which require materials having relatively long procurement cycles.

Detailed manufacturing drawings have been completed on the following parts and assemblies:

a. Stage 1 and 2 buckets.

b. Stage 1 and 2 discs.

c. Stage 1 and 2 nozzle diaphragms and the outlet guide vane assembly with all necessary instrumentation.

d. Casing.

e. Bucket tip shrouds.

f. Interstage seals.

g. Potassium slinger seal.

h. Exhaust scroll.

i. Pivoted pad bearing details and assembly.

The following drawings are as yet incomplete:

a. Shaft.

b. Shaft support cone structure.

c. Turbine inlet section.

In addition to the above detailed manufacturing drawings, drawings showing axial and radial stack-up tolerance errors and differential thermal expansion between both rotor and stator are being made to insure adequate clearance between rotating and stationary parts during operation. Provisions for shims have been made to permit necessary adjustments during assembly. The flow path has been drawn showing dimensions at room and operating temperatures.

The complex intersection of the twisted nozzle vane with the conical inner and outer shroud bands of both nozzle diaphragm assemblies has been determined and glassines have been made of the necessary tooling which must be used for the electrical machining of the vane slots in the shroud bands.

5. <u>Stress Analysis</u>

Stress analyses are being conducted on both rotating and stationary parts. On the stationary parts, in addition to the casing stresses discussed previously, an investigation has been made of the stresses in both of the nozzle diaphragm assemblies. Because 316 stainless steel has been chosen for the casing material, as previously discussed, it has been decided to manufacture the inner and outer shroud bands of both nozzle diaphragm assemblies and the outlet guide vane from 316 stainless steel also. Since these parts must be rabbetted into the casing, this material choice permits the differential thermal expansion between the shroud bands and the casing to be minimized. The nozzle vanes of both stages 1 and 2 are L-605 and the outlet guide vanes are 316 stainless steel. Because of the lower allowable stresses of 316 stainless steel at the high temperatures involved, it is important to correctly and thoroughly analyze the nozzle diaphragm stresses and identify any potential stress problems. Each nozzle diaphragm assembly sustains an axial pressure force in excess of 600 lbs. By using relatively heavy sheet stock, and box sections where possible, and by utilizing appropriate shapes of inner bands, all calculated stress levels have been held to less than the allowable stress corresponding 0.2 per cent creep strength.

The final stress analysis of the rotor blades incorporating the latest fluid dynamic changes is being completed, and indications are that while the bending and centrifugal stresses of the airfoil root sections are well within bounds, the wheel dovetail stresses are approaching the limits imposed by the selected failure criteria. In addition to the stress analysis, the criteria for allowable stresses is presently being scrutinized in the light of turbojet engine operating experience. As previously indicated, the stresses in the bucket dovetails have increased due to the increased weight of the buckets. In the final design the highest stresses compared to the allowable levels occur in the disc dovetail slots of the first stage and in the slots of the second stage which carry refractory alloy buckets.

In addition to the blade stress analysis, the tie-bolt stress analysis is being completed. As reported previously, the tie bolt is stress limited at its hot end where it protrudes through the first stage wheel and as the temperature rapidly drops along its length the allowable stress increases proportionately. Because of limitations on its diameter and stress level at the hot end, the maximum allowable force which the tie bolt can sustain when in tension and yet remain within the 0.2 per cent creep limits at its hot end is in the neighborhood of 10,000 lbs. The bolt has been designed to take advantage of the higher allowable stress in its cooler sections by reducing its diameter and thereby increasing stress as the temperature drop allows. The bolt diameter is reduced in two steps from the hot end to the cold end. The values of diameter are: 0.85, 0.7, and 0.6 inches. By thus reducing the bolt diameter, the stress is increased along the bolt, resulting in increased bolt elongation. The total elongation is approximately 0.023 inches while the calculations show the differential thermal expansion between tie-bolt and shaft to be about 0.002 inches between cold and hot conditions. The increased elongation of the bolt will allow for considerable error in expansion calculations without the danger of the bolt loosening or stressing the hot end beyond its limits under operation. A thermal analysis has been made of the assembled turbine from which the relative positions of the cold rotor and stator have been established such that, after thermal expansion, the rotor-stator minimum clearances will be the same as those experienced while the turbine is cold.

The bucket vibration analysis is under way using the final airfoil sections. Because of the high natural frequencies characteristic of the short, stiff buckets, it appears that the aerodynamic excitations due to struts, etc. upstream of the turbine will not be of high enough frequency to cause problems of bucket resonance. However, a preliminary investigation shows that the excitation frequencies produced by the buckets alternatively passing in and out of the nozzle jets are close enough to the bucket natural frequencies to be of concern.

Due to the high stiffness of the buckets, the buckets cannot be analyzed as a twisted beam rigidly mounted at its airfoil root, but rather the springiness of the dovetail - wheel attachment must be considered. Previous experience indicates that in buckets of similar geometry, flexural natural frequencies have been measured which are as much as 50 per cent below those obtained by analyses which ignore the attachment flexibility. The present analysis is being done on existing computer programs, but has not yet progressed to the point of accurately defining the bucket natural frequencies. Ultimately, despite the present analysis, the bucket resonance points will have to be accurately checked by testing when the manufactured parts are available, and if necessary, the turbine operating speeds will have to be adjusted to avoid them.

6. Maintenance Provisions

An extensive investigation is being made into the problems of either inspecting or repairing the turbine parts at various times during the test program. It is known that opening the turbine casing after a period of operation poses a contamination problem due to the oxidation of the potassium remaining on the surfaces of the inner parts of the turbine. For this reason, consideration has been given to enclosing the turbine in a glove box, and keeping this container filled with argon both during system operation and during turbine inspection. A detailed design of this component is under way. Shown in Figure 14 is a preliminary layout of the glove box. With an inert gas cover, the turbine casing could be opened for the inspection of blades, instrumentation, etc., and at least some minor turbine parts could be removed if necessary without contaminating the test turbine or the facility. The glove box is being designed for maximum accessibility and part replacement, and the extent to which overhaul can be accomplished within the glove box is being investigated. For further safety against contamination of the facility, an investigation is being made into the capping off of the facility also during inspection or overhaul of the turbine. The requisite changes in the turbine mechanical design to facilitate turbine maintenance in the glove box are being determined, and the design is being modified to incorporate these changes.

If it should become necessary to remove the entire turbine from the facility at any time, the exposure to the atmosphere would require that before returning it to the facility all parts would have to be cleaned thoroughly to remove oxides, and at least the turbine would have to be purged thoroughly upon reinstallation.

C. TESTING PROGRAMS

In parallel with the detailed design programs, various individual testing programs are under way to check the validity of design decisions regarding fabrication and material properties in time to make any necessary design changes.

As reported in the section on MATERIALS SUPPORT, the testing of astroloy samples is under way to verify its reported strength properties.

Because of the hardness of astroloy, the broaching of the dovetail slots in the wheels may be a problem. Since broaching is a difficult operation on hard materials, it may be necessary to do this machining operation while the material is not yet in its fully heat treated and hardened condition. An investigation is being made into broaching dovetail slots at an intermediate heat treatment condition and then aging the material to its full strength and hardness. A test has been devised wherein six coupons of astroloy, divided into three pairs, have been given three different heat treatments, each representing the wheel material at some phase of its heat treatment cycle. One pair has been heat treated to obtain minimum hardness, one pair for minimum grain size, and one pair having all but the final aging heat treatment. All coupons were broached, and machined well with no deleterious effects on the quality of the broached slot or the broaching cutter. Each coupon will subsequently be heat treated to the final treatment proposed for the astroloy wheels. The dimensions of the slot in each coupon will be accurately checked both before and after its heat treatment, to determine whether the dimensional stability of the material during heat treatment prevents unacceptable warping of the dovetail slot.

A test sample has been made to determine the integrity of a 316 stainless steel - L-605 brazed joint characteristic of that between the L-605 nozzle partitions and the 316 stainless steel shroud bands in the nozzle diaphragms. The test sample is shown in Figure 15. An L-605 tube and solid rod were brazed into a 316 stainless steel container, severely testing the resistance to cracking of the H-33 brazed joint under differential thermal expansion. The braze flowed very well, made very adequate fillets, and did not crack, indicating that brazing rather than welding of the nozzle diaphragm assemblies is a satisfactory manufacturing method. Several microphotographs were made to bear out the quality of the joint.

As shown in Figure 10, bolts have been used in the assembly of the turbine tip shrouds to the casing. In the presence of an alkali metal atmosphere, it is feared that uncoated R-41 bolts will diffusion-bond and thereby seize with the 316 stainless steel during operation, making replacement of these turbine parts difficult. Since the use of bolts cannot be eliminated, a test program is under way to find possible coatings, which when applied to the bolt, will not be washed away by the potassium, and will facilitate bolt removal. Capsule tests of R-41 bolts inserted in tapped holes in 316 stainless steel coupons in the presence of hot potassium will be run.

In a related sense, due to the similarity of the turbine wheel and bucket materials, a potential fretting problem exists. It is being investigated, and, if necessary, tests will be conducted when the requisite parts become available.

D. PROCUREMENT

In general, turbine hardware procurement has been initiated to the extent of obtaining manufacturing cost quotations from various vendors on all components represented by the previously listed completed manufacturing drawings.

One important phase of procurement regards the compilation of manufacturing and metallurgical specifications for turbine components. On all components represented by completed manufacturing drawings, detailed specifications have been defined for the procurement of material which will meet the material strength and alkali metal compatibility requirements. Specifications for sheet, plate, strip, bar stock, and forgings have been defined for various materials. These largely require modification of existing AMS, ASTM, or contractor specifications, by the inclusion of more elaborate ultrasonic testing, high strength requirements, and more stringent grain size requirements (due to the small size of the turbine).

An equally important area which has been given extensive attention is in regard to the detailed drawing notes. These have been written so as to clearly delineate in sequence the required manufacturing processes to conform to contractor specifications and to the special requirements of the alkali metal atmosphere. Clearly defined, step-by-step, manufacturing procedures with associated heat treatments are especially required with such delicate, high precision parts as the pivoted pad bearing, for example, where processes of machining, heat treatment, surface hardening, and surface plating are all required at one or more times in its manufacture.

E. INSTRUMENTATION

The instrumentation required by the potassium test turbine will be in two groupings, that required during the pretesting in steam atmosphere and that required by the turbine during its performance and endurance runs in potassium.

During the pretesting phase, the turbine instrumentation will be aimed toward a check-out of the mechanical functioning of the turbine parts rather than performance data. As indicated previously, because of the high rotor stress levels and proximity of resonant frequencies to the operating speeds, accurate measurements will have to be made both of rotor stresses and blade natural frequencies in order to select operating speeds which do not interfere with these limitations. The design of strain gauge instrumentation on the turbine buckets and wheels is presently under way, to determine both the number and the location required to obtain the necessary mechanical data.

The turbine performance instrumentation requirements for the potassium test turbine, exclusive of facility instrumentation, has been established and is summarized in Table III. Shown in Figure 16 is a schematic diagram of the test turbine showing instrumentation stations. There are eight instrumentation stations disposed between the station upstream of the liquid injection station and the one downstream of the turbine exit scroll. Two additional stations are associated with the two moisture extraction stations. Shown in Figure 17 are schematic diagrams of the pressure and temperature sensor locations at each station. Stations 1 and 2 permit a heat balance to be made so that the resulting quality after injection may be determined. Stations 3 and 7 provide turbine inlet and exit quantities used in the efficiency determination. At these stations, static and total pressure measurements will be made at circumferential locations approaching 180 degrees apart, providing means of determining the existance of axisymmetry. At the intra-and interstage stations 4, 5 and 6, static pressures on one side of the casing only will be taken. Station 8 data will be used to relate the quality measurement at this station back to the turbine exit station (Station No. 7).

It is necessary to know the amount of argon entering the test facility from the efflux pressure measuring devices because the argon contains traces of oxygen which accelerates corrosion. Provisions must be made for removing this oxygen at the rate it enters to prevent a build-up of contaminants. Shown in Figure 18 is the experimental variation of the argon flow per measuring device as a function of manifold pressure. A manifold pressure of 90 psia has been selected for the efflux system resulting in a total argon flow of 5.1×10^{-4} pps for the 36 devices to be used on the test turbine. This amounts to about 0.02 per cent of the design vapor flow rate of the turbine.

During anticipated pressure surges in the testing of the potassium turbine or to remove potassium vapor from the pressure leads a purge gas flow can be actuated. Shown in Figure 19 is the experimental variation of this flow with manifold pressure. For a purge-cycle manifold pressure of 175 psia the total argon flow through the 36 pressure leads is 1.67×10^{-2} pps. This amounts to 0.65 per cent of the design vapor flow rate and indicates that the purge flow must be used sparingly to keep from overloading the facility oxygen removal equipment.

IV. TEST FACILITY

A. TEST FACILITY DESIGN

As pointed out last quarter, the liquid metal facility required for testing the two-stage potassium turbine is to operate on the Rankine cycle. A 2.8 lb/sec potassium vapor flow at $1600^{\circ}F$, turbine inlet, is required. The basic components necessary for Rankine cycle operation are a boiler, turbine, condenser, and feed pump. An extensive study was made to optimize the arrangement and method for mounting these components. Simplicity, reliability, minimum building modification and minimum piping stress resulting from thermal growth were predominate criteria for the loop arrangement.

All loop arrangement criteria have been met with good success except for minimum building modification. The initial approach was to locate the test loop in the existing Space Projects Laboratory (Building 309). Due primarily to the large size of the test loop, it has been concluded that it will be more feasible and economical to construct a new wing to the existing Space Projects Laboratory for housing the test loop. Design work has been started on the building addition, and construction is expected to be started within 90 days.

Due to its large size and weight, in all layout studies the boiler has been assumed fixed. The loop layout which has the most desirable features is shown diagramatically in Figure 12. The geometric center-lines of all major loop components are in the same vertical plane. Thermal expansion of the loop relative to the boiler is permitted through the use of rollers or flexible constant force hangers. The potassium vapor line is a straight run from the boiler vapor drum. The test turbine, turbine dynamometer, and system condenser are mounted on a low friction movable dolly which absorbs the axial thermal expansion (about 2-1/2 inches) of the hot vapor pipe. The boiler liquid feed line will be sufficiently flexible for absorbing relative thermal expansion.

Although not exact in all details, Figure 20 is a representative isometric view of the test loop general arrangement. This illustration accurately presents the relative size of the test loop.

Several bids have been received for the fabrication of the boiler and condenser. Price and delivery have been quoted for a boiler made of L-605 tubes and one made of type 316 stainless steel tubes. Based primarily on cost of delivery and some uncertainty of boiler performance, a decision has been made to have the boiler tube bundle fabricated of 316 stainless steel with provisions for an L-605 replacement tube bundle when required. Also no attempt will be made to super-heat the potassium vapor in the 316 stainless steel system. Several boiler manufacturers have expressed assurance that 98 to 99 per cent vapor quality can be obtained by proper separation in the vapor drum. The 316 stainless steel tube bundle allows an upper limit of 1650°F for the metal wall temperature, which insures 1600 to 1620°F vapor temperature. This temperature and corresponding saturation vapor pressure capability is adequate to completely satisfy the requirements of the present turbine test program.

Although fundamentally the same, five different boiler designs have been given an intensive review. Reliability, price and delivery are considerations for boiler selection, but the major criterion is that the boiler be technically acceptable for liquid metal boiling to the best of our knowledge. Settlement on design details for the boiler is nearly completed, and a purchase order is expected to be written within a few weeks.

Bids have also been received for the test loop condenser. Much effort has been applied to the design of the condenser to insure that torsional, bending and axial loads are not imposed on the test turbine scroll through the rigid vapor pipe connecting the turbine to the condenser. The condenser is air cooled. Consequently, the condenser tube bundle must be designed so that both aerodynamic drag forces and gravitational forces on the tube bundle are counter-balanced so that the turbine exhaust pipe is not subjected to a significant resultant force.

An inleakage of 0.0005 lb/sec of argon at the test turbine is anticipated. This represents 0.02 per cent of full load vapor flow. Methods for removing and reclaiming this argon from the condenser are being studied. Also a functional comparison of a vertical tube condenser with a horizontal tube condenser is being made. Design details have been nearly completed. A purchase order for the test loop condenser is expected to be issued within a few weeks.

Failure of the test turbine, the load absorber or loss of speed control may require an emergency shutdown of the test loop. The simplest and most positive method for isolating the test turbine from the test loop is by closing a quick operating valve in the turbine inlet and outlet vapor lines. Boiler shutdown would be initiated simultaneously with closing of the valve to prevent a dangerous over-pressure of the boiler. Valve specifications have been prepared, and several companies have been contacted to discuss detail valve design. Since the valve maximum operating temperature is 1600°F, butterfly type valves are recommended. Bids for fabrication of these valves are expected from several manufacturers within a few weeks.

Several methods for measuring vapor quality are being considered. These are gamma or X-ray densitometers and a heat calorimeter. Radiation densitometers do not seem to have sufficient sensitivity and/or adaptability for this particular application. The heating calorimeter concept consists of an electrical heating device, temperature and pressure gages and flowmeter. A continuous vapor sample is superheated so that its enthalpy may be determined from temperature and pressure measurements. Knowing the electrical heat input and vapor sample weight flow, the initial enthalpy and, consequently, the initial quality of the sample can be determined from the Mollier diagram. Accuracies in the order of 1 per cent are expected with this system if a truly representative vapor sample can be extracted.

B. POWER BRAKE

Several power absorbing systems have been investigated and found to be suitable, within limits, for loading the potassium test turbine. Eddy current brakes, air brakes and water brakes are receiving a certain amount of attention. No one brake or braking system has conclusively met all of the desired characteristics involving cost, size, availability, simplicity and reliability.

The first choice for the power absorbing device is a surplus eddy current dynamometer. This unit is rated for 3000 hp from 6400 rpm to

24,000 rpm. Its physical size and weight are quite large, which, although presenting some mounting problems, do not restrict its usefulness for its intended application. Only minor modifications are required to adapt this unit to the test turbine.

An investigation of the adaptability of an aircraft turbo-supercharger as a load absorber for the potassium turbine has continued. Both open and closed cycle systems have been analyzed for Models BH4 and B31 turbo-superchargers. Open cycle operation for either unit becomes marginal for full range of power and speed. Even though either supercharger can be made to load the turbine over the full range of speed and power conditions in a closed cycle, the B31, the smaller unit, is better suited for this application. Two B33 superchargers (very similar to B31) have been received in a completely overhauled condition. Cost estimates have been prepared for both the open cycle and closed cycle turbo-supercharger load absorber. Although the initial purchase price of the basic turbo-supercharger is low, the cost of a closed cycle system is nearly that of a new eddy current brake. The open cycle, although lacking in full range load absorbing capacity, is appreciably less expensive. An improvement of open cycle performance can be made by means of inlet guide vanes.

The use of a water brake has been investigated further. From past experience with rotating disk type water brakes, cavitation and erosion are so severe that brake life is limited to 150 to 200 hours. The Hydro-Mill Company has developed a vortex type water brake in which they claim cavitation does not occur, thus, theoretically, extending brake life in excess of 1000 hours. Their brake is a recent development and no experimental data are available to support this claim. The attractive feature of this brake is its extremely small size and low cost. At present no action is being taken to verify the acceptability of this type brake.

V. MATERIALS SUPPORT

During the reporting period, the materials selection was reviewed, and specifications were prepared for the procurement of the turbine and test facility materials. The TZM and F-48 refractory alloy turbine blade materials were obtained, and their evaluation is in progress. Several experimental investigations were initiated to examine fabrication procedures and to obtain more extensive documentation of some of the materials which will be used in the turbine.

A. MATERIALS SELECTION

The major turbine components for which material selections have been made are listed below.

	Component	Material Selection	Max. Temp., °F
1.	Stage One Disc	Astroloy	1495
2.	Stage Two Disc	Astroloy	1375
3.	Stage One Blades	U-700	1415
4.	Stage Two Blades	U-700	1285
5.	Turbine Casing	Type 316 SS	1600
6.	Nozzle Partitions	L-605	1600
7.	Aft Casing	Type 316 SS	1365
8.	Tie Bolt	U-700	1500
9.	Shaft	A-286	1100

During the last quarter, the previous selection of L-605 for the turbine casing parts was changed to Type 316 stainless steel. This will conform to the use of stainless steel for the piping preceding and following the turbine. The strength of the stainless steel is considered adequate for design purposes, and its use has the benefits of (a) avoiding bimetallic joints in the piping, (b) a somewhat superior aging-embrittlement behavior, and (c) a lower cost. For the nozzle partitions, L-605 is still preferred because of its higher strength and possibly superior erosion resistance. The required joining of L-605 partitions to stainless steel diaphragm bands can be accomplished either by welding (Reference 1) or by brazing with the H-33 cobalt base alloy which has exhibited good corrosion resistance to potassium at temperatures up to 1850°F (Reference 1). It is presently planned to use the brazing method in expectation of maintaining better dimensional control.

The previous selection of U-500 for the tie bolt has been changed to U-700. This change has been made primarily with regard to the decrease in overall cost which can result from combining the turbine blade and tie bolt materials order to purchase a larger quantity of U-700. The superior strength of U-700 is also expected to be an advantage.

Standard specifications for the procurement and fabrication of the selected materials were reviewed, and appropriate modifications to these specifications were largely completed during the reporting period.

B. FABRICATION PROCEDURES

1. Broaching of Astroloy Discs

The use of Astroloy for the turbine discs presents a potential problem in the broaching of narrow dovetail slots because of the high hardness of the fully heat treated material. This is evident from the facts that:

a. From a manufacturing standpoint, a relatively low hardness is desirable for broaching.

b. From an engineering standpoint, higher tolerances are attainable if broaching is conducted after all heat treatments, which implies a high hardness.

In order to identify a fabrication sequence which best satisfies both the manufacturing and the engineering requirements, a series of broaching trials is in progress, using Astroloy specimens which have been heat treated to various conditions. The heat treatments have been completed and the results are shown in Table IV.

Apparently, the extra two hours at $2140^{o}F$ in treatments (a) and (c) does not significantly increase the average grain size, but does increase the hardness, possibly by dissolving carbides. Because of the small dimensions of the dovetail prongs, it is important and necessary that the increased time at the solution annealing temperature does not greatly increase the grain size (decrease the ASTM Grain Size No.).

The broaching has been completed, and evaluation of the specimens is in progress.

2. Brazing of L-605 to Type 316 Stainless Steel

It is planned to braze the L-605 nozzle partitions to the Type 316 stainless steel diaphragm bands. In a brazing trial, a mock-up of the joint was constructed, brazed, and evaluated. The H-33 brazing alloy (21 Ni, 21 Cr, 8 Si, 3.5 W, 0.4 C, 0.8 B, bal. Co) was used in conjunction with the following brazing cycle in hydrogen.

a. Total heating time - 30 minutes

b. Time at 2150^{o}F - 15 minutes.

The sectioned test piece is shown in Figure 21, and the microstructure of one of the brazed joints is shown in Figure 22. Metallographic examination indicated that the joint was sound, and fluorescent penetrant inspection indicated that there were no cracks in the joint.

C. MATERIALS EVALUATION

The materials evaluation work is being conducted to document some of the strength-ductility characteristics of critical turbine materials and to examine the corrosion resistance of critical materials to environments which, in some respects, simulate expected operating conditions.

1. <u>Strength-Ductility Characteristics</u>

The rotating parts of the turbine will encounter combinations of temperatures, times, and stresses which will approach the creep limits of the strongest available superalloys, i.e., Astroloy and U-700. The tests in progress will provide a comparison of the room temperature tensile and creep rupture properties of U-700 specimens heated in potassium and in air. In addition, the three refractory alloys to be evaluated in the turbine will be documented with respect to tensile and stress-rupture properties under similar conditions.

The results obtained to date are for material as received from the vendors, plus solutioning and aging heat treatments. The U-700 data is listed in Table V, and Table VI contains the data for the TZM turbine blade stock.

Although the ductility and rupture life of this U-700 test material is somewhat below that desired and specified for the material to be used in the turbine (Table V), it is not expected that this will greatly detract from the results obtained from the experiments in progress. Creep testing of the as-heat treated material is in progress, and tensile and creep-rupture testing of specimens exposed to potassium and air will be initiated during the next quarter.

The TZM turbine blade material exhibits somewhat lower tensile strength and higher ductility than is sometimes reported for this alloy. However, these properties are believed adequate for the present application.

2. Oxidation of Type 316 Stainless Steel

The oxidation of Type 316 stainless steel is of interest with respect to the use of this material for the potassium heater at temperatures up to 1650°F. Because the available data are quite limited in quantity and not directly applicable to the expected service conditions, some oxidation tests have been initiated. In one set of experiments, specimens are being heated at 1550° and 1650°F for 1,000 hours in still air, with intermittant thermal cycles to room temperature. It is planned to conduct another set of tests in a slightly oxidizing combustion gas environment.

3. The Effect of Potassium on U-700

Exposures of U-700 to potassium for 100 and 1,000 hours at 1500°F have been completed. The specimens, 0.5 inch diameter cylinders about 1.1 inches long, were contained in 1.165 inch O.D., 0.063 inch wall L-605 capsules, filled with about 28 grams of slagged, filtered, distilled, and hot trapped potassium which was purchased to a specification requiring less than 50 ppm of oxygen. (This is the same

Type A potassium which has been used for other corrosion investigations (Reference 1).) The U-700 thickness decrease after the 100 and 1,000 hour exposures was 0.1 and 0.5 mil per side, respectively. The metal loss is accompanied by the formation of a loosely adherent scale. In addition, a surface layer of less than 0.1 mil appears to have undergone alloying element depletion or contamination (Figure 23). No intergranular attack was observed. Additional tensile and rupture bar specimens are being exposed to potassium for subsequent determination of the U-700 mechanical properties.

4. Evaluation of Alumina Coatings on F-48

To evaluate the possibility of using a coating for erosion protection of the turbine blades, specimens of F-48 were flame sprayed with alumina and ground so that two levels of coating thickness were obtained, i.e., 4 mils and 6 mils. Metallographic examination indicates that there are very few interconnected pores to the base material. Because it would be extremely difficult to spray the alumina over the thin trailing edge of the blade, only the area of expected erosion will be coated, i.e., the convex surface of the leading edge. To simulate this condition, the ends of the specimens were beveled through the F-48 base metal. The specimens were sealed in a Cb-1Zr capsule filled with slagged, filtered, distilled, and hot trapped potassium. This capsule was then sealed in an L-605 capsule for oxidation protection and is being exposed at 1400°F for 500 hours with 10 cycles to room temperature.

D. PROCUREMENT OF THE REFRACTORY ALLOYS

The status of the refractory alloy procurement for preparation of the turbine blades is as follows:

1. TZM Molybdenum Alloy

Twelve feet of 1.25 inch diameter bar of molybdenum alloy TZM has been received from the Climax Molybdenum Company (Heat No. TZ-6006). This will provide blanks for 30 turbine blades and material for experimental evaluation, with four feet of excess bar which can be used for additional blade blanks or evaluation if necessary. The material is in the stress relieved condition (2300°F), with approximately 75 per cent reduction in area following the last recrystallization heat treatment.

The quality of the material appears very good, as determined by ultrasonic and fluorescent penetrant inspection, metallographic examination, hardness tests, and chemical analyses. Strength and ductility characteristics of the material are reported in Table VI.

2. AS-30 Columbium Alloy

Three electrodes, 2.25 inch diameter and 18 inches long, were obtained from Kennametal Inc. They were subsequently melted at the General Electric Company Research Laboratory into a 5.5 inch diameter crucible. The ingot was conditioned and sealed within a 5 inch diameter, 0.125 inch wall molybdenum can. Extrusion was successfully accomplished at 1800^{o}C with a 5.8-to-1 reduction ratio to a 2.07 inch diameter billet. After conditioning, the billet was sectioned into 3.5 inch lengths which will be re-extruded to 1.1 x 1.35 inch cross section bars. Double extrusion is being used in an effort to obtain a fine grain size.

3. F-48 Columbium Alloy

A 0.75 inch thick plate 10.5 x 17.75 inches was received from the Crucible Steel Company of America (ASD Cb sheet rolling program Contract AF 33(600)-39942). Details of the processing will be supplied by Crucible Steel Company and reported at a later date. The material will be inspected by X-ray, ultrasonic, and fluorescent penetrant techniques in addition to metallographic examination, hardness tests, chemical analyses, and mechanical property determinations.

E. REFERENCE

1. Alkali Metals Boiling and Condensing Investigations, Quarterly Reports for Contract NAS 5-681 (1961).

TABLE I

COMPARISON OF DESIGN DATA FROM PRELIMINARY AND FINAL BUCKET DESIGNS

Stage	1		2	
Configuration	Prelim.	Final	Prelim.	Final
Trailing edge overhang	Yes	No	Yes	No
Dovetail angle, deg.	5.25	7.75	2.75	7.50
Bent platform	No	No	Yes	No
Hub area, sq. in.	.1020	.1043	.1113	.1215
Pitch area, sq. in.	.0847	.0835	.0947	.0998
Tip area, sq. in.	.0630	.0630	.0771	.0771
Centrifugal load on bucket, lb.	1408	1386	1908	2068
Clearance for replacement, in.	None	.029	None	.046

TABLE II

PHYSICAL DIMENSIONS OF FINAL TEST TURBINE BUCKETS

Stage	1			2		
Radial Location	Tip	Pitch	Hub	Tip	Pitch	Hub
Radius	4.40	4.08	3.76	4.80	4.40	4.00
Blade Inlet Angle, Deg.	59.0	62.4	65.3	52.3	57.3	62.0
Blade Exit Angle, Deg.	64.6	66.2	63.0	62.6	64.8	64.0
Max. Thickness, In.	.137	.165	.190	.1475	.1735	.194
Aerodynamic Chord, In.	.618	.646	.682	.724	.752	.789
Axial Width, In.	.584	.6345	.684	.6815	.736	.789
Stagger Angle, Deg.	22.0	15.2	8.9	22.5	15.5	9.5

(Measured from line tangential to blade)

TABLE III

POTASSIUM TEST TURBINE PERFORMANCE INSTRUMENTATION

Item	Station No.	Location	Quantity	No.	Max. Value *
1	1	Upstream Liquid Inject. Sta.	Vapor T. Temp.	5	$1650^{o}F$
2		Upstream Liquid Inject. Sta.	Vapor T. Press.	5	45 psia
3		Upstream Liquid Inject. Sta.	Vapor S. Press.	2	45 psia
4		Upstream Liquid Inject. Sta.	Liquid Temp.	1	$1650^{o}F$
5		Upstream Liquid Inject. Sta.	Liquid Press. **	1	150 psia
6		Upstream Liquid Inject. Sta.	Vapor Flow	1	3 pps
7		Upstream Liquid Inject. Sta.	Liquid Flow	1	0. 5 pps
8		Main Shaft	Rotative Speed	1	24, 000 rpm
9		Main Shaft	Torque	1	120 ft-lb
10	2	Upstream Bullet Nose	Vapor Temp.	5	$1600^{o}F$
11		Upstream Bullet Nose	Moisture Fract.	1	15%
12	3	Upstream Nozzle No. 1	Vapor T. Press.	4	38 psia
13		Upstream Nozzle No. 1	Vapor S. Press.	4	38 psia
14	4	Upstream Rotor No. 1	Vapor S. Press.	2	26 psia
15		Wheel Space Rotor No. 1	Vapor Temp.	2	$1500^{o}F$
16	5	Upstream Nozzle No. 2	Vapor S. Press.	2	22 psia
17		Wheel Space Nozzle No. 2	Vapor Temp.	2	$1450^{o}F$
18	6	Upstream Rotor No. 2	Vapor S. Press.	2	15 psia
19		Wheel Space Rotor No. 2	Vapor Temp.	2	$1400^{o}F$

* Tabulated values not all pertinent to same condition of operation.
** Taylor gage.

POTASSIUM TEST TURBINE PERFORMANCE INSTRUMENTATION (cont'd)

Item	Station No.	Location	Quantity	No.	Max. Value *
20	7	Upstream OGV	Vapor T. Press.	4	13 psia
21		Upstream OGV	Vapor S. Press.	4	12 psia
22		Upstream OGV	Vapor T. Temp.	6	1300°F
23	8	Downstream Scroll	Vapor T. Temp.	6	1300°F
24		Downstream Scroll	Vapor T. Press.	5	10 psia
25		Downstream Scroll	Vapor S. Press.	2	10 psia
26		Downstream Scroll	Moisture Fraction	1	20%
27	9	Removal Tank Stage No. 1	Metal Temp.	2	1500°F
28		Removal Tank Stage No. 1	Liquid Flow	2	0.25 pps
29		Removal Line Stage No. 1	Liquid Temp.	1	1500°F
30	10	Removal Tank Stage No. 2	Metal Temp.	2	1400°F
31		Removal Tank Stage No. 2	Liquid Flow	2	0.25 pps
32		Removal Line Stage No. 2.	Liquid Temp.	1	1400°F

* Tabulated values not all pertinent to same condition of operation.

** Taylor gage.

TABLE IV

HEAT TREATMENT OF U-700 FOR BROACHING TRIALS

	Heat Treatments	Hardness, Rc	ASTM Grain Size No.
a.	4 hours @ 2140°F, A.C.	40.0	2.1
	4 hours @ 1975°F, A.C.	37.4	2.6
	8 hours @ 1550°F, A.C.	38.7	(2.6)
b.	2 hours @ 2140°F, A.C.	38.3	2.8
	4 hours @ 1975°F, A.C.	36.0	2.3
	8 hours @ 1550°F, A.C.	39.3	(2.3)
c.	4 hours @ 2140°F, A.C.	40.0	2.1
	4 hours @ 1975°F, A.C.	37.4	2.6

Final aging at 1400°F will be conducted after broaching in all cases.

TABLE V

U-700 MECHANICAL PROPERTIES

Heat No. 42555

Chemistry - 18.7% Co | 4.48% Al | 0.029% B | <0.05% Zr | 0.10% Fe
15.1% Cr | 3.40% Ti | 0.06% C | 5.15% Mo

Test Temp.	Ultimate Tensile* Strength, psi	0.02% Yield Strength, psi	Elongation in 1", %	Reduction in Area, %
R. T.	157,000	134,500	5.4	7.5
R. T.	170,300	138,000	7.1	6.5
Turbine Specification (R. T.)	(150,000)	(130,000)	(12)	

Test Temp.	Stress, psi	Time, Hrs.	Elongation in 1", %
1400°F	85,000	21.7	4.4
1400°F	85,000	46.0	4.5
Turbine Specification (1400°F)	(85,000)	(30)	(5)

TABLE VI

TZM MECHANICAL PROPERTIES

Heat No. TZ-6006

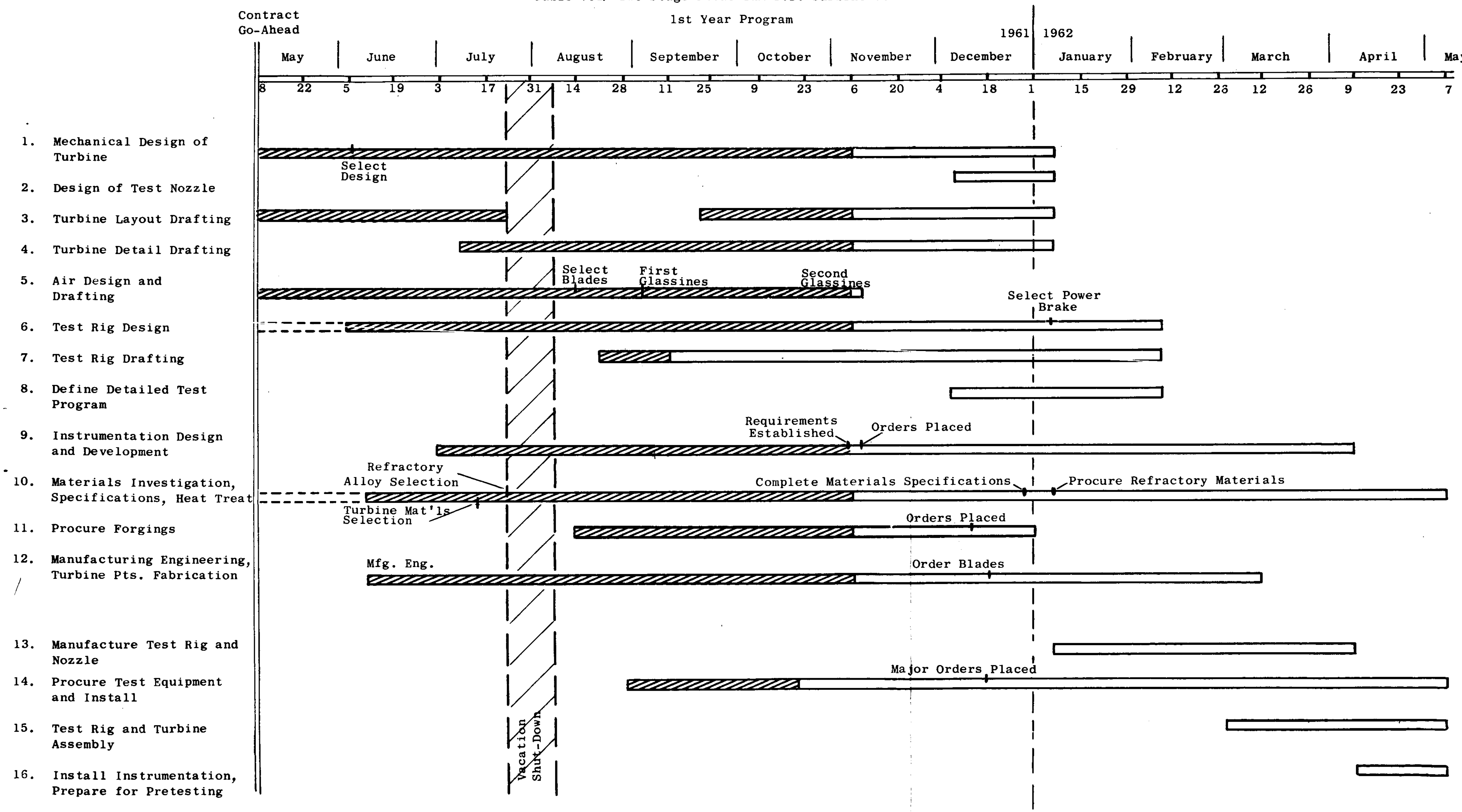
Chemistry - 0.012% C | 0.09% Zr | 0.002 max. %N
0.55% Ti | 0.0025 max. %O

Test Temp.	Ultimate Tensile* Strength, psi	0.02% Yield Strength, psi	Elongation in 1", %	Reduction in Area, %
R. T.	108,200	99,600	26.1	62.7
R. T.	118,100	104,000	21.3	54.8
1400°F	87,600	80,000	13.4	83.5
1400°F	87,700	81,400	14.3	83.0

* Strain rate = 0.005"/"/min to yield.

Table VII. Two Stage Potassium Test Turbine Schedule

1st Year Program



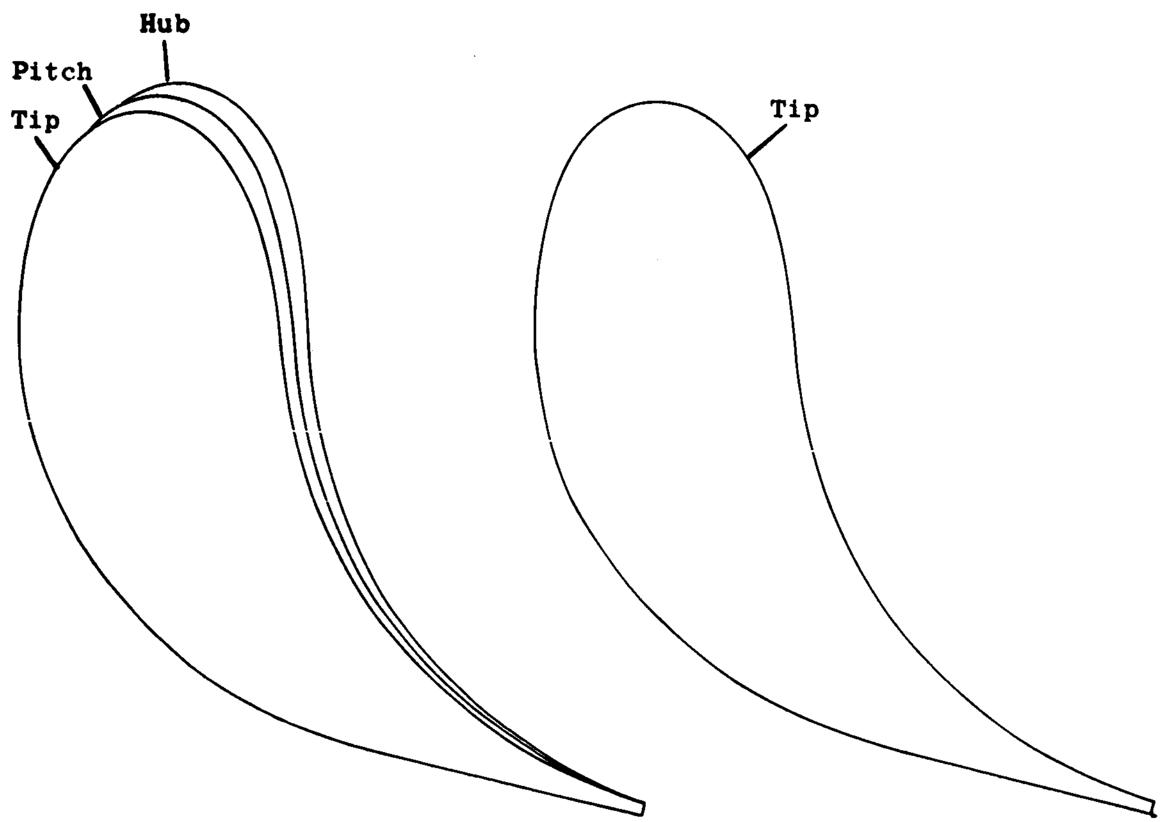


Figure 1. **Final First Stage Nozzle Partition Section Profiles**

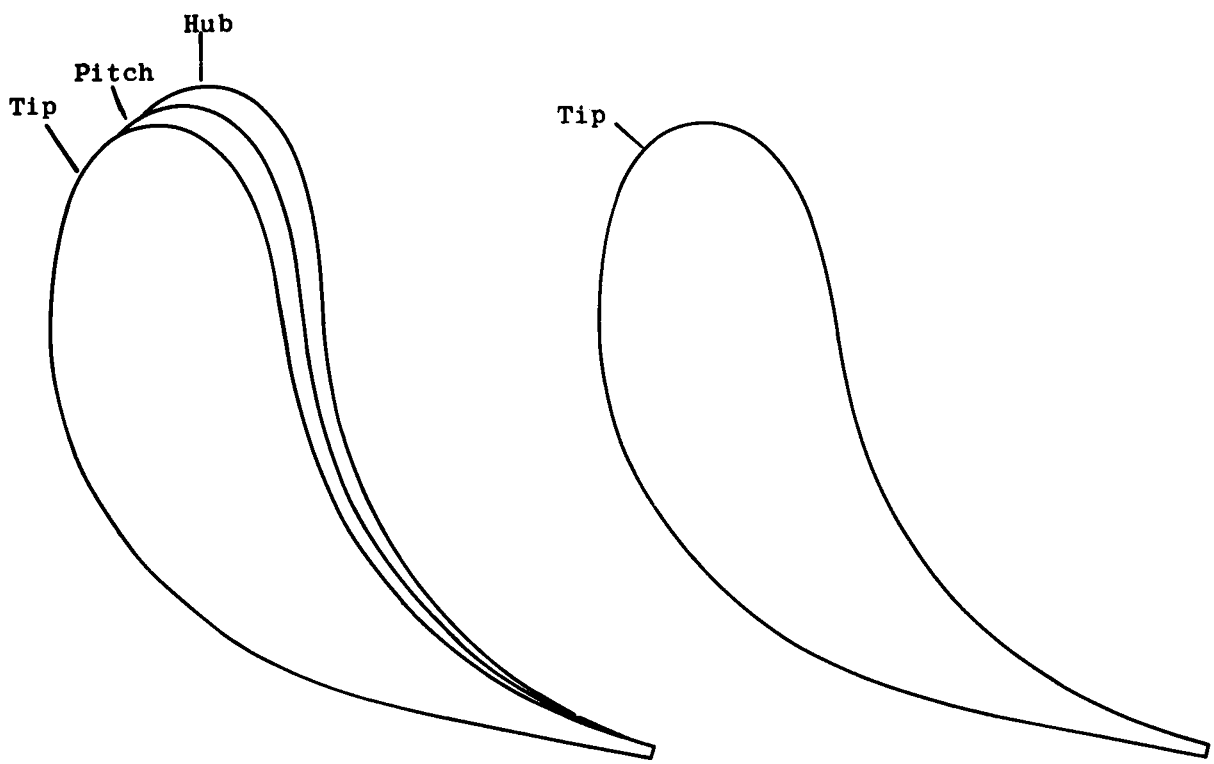


Figure 2. Final Second Stage Nozzle Partition Section Profiles

(a) Preliminary

(b) Final

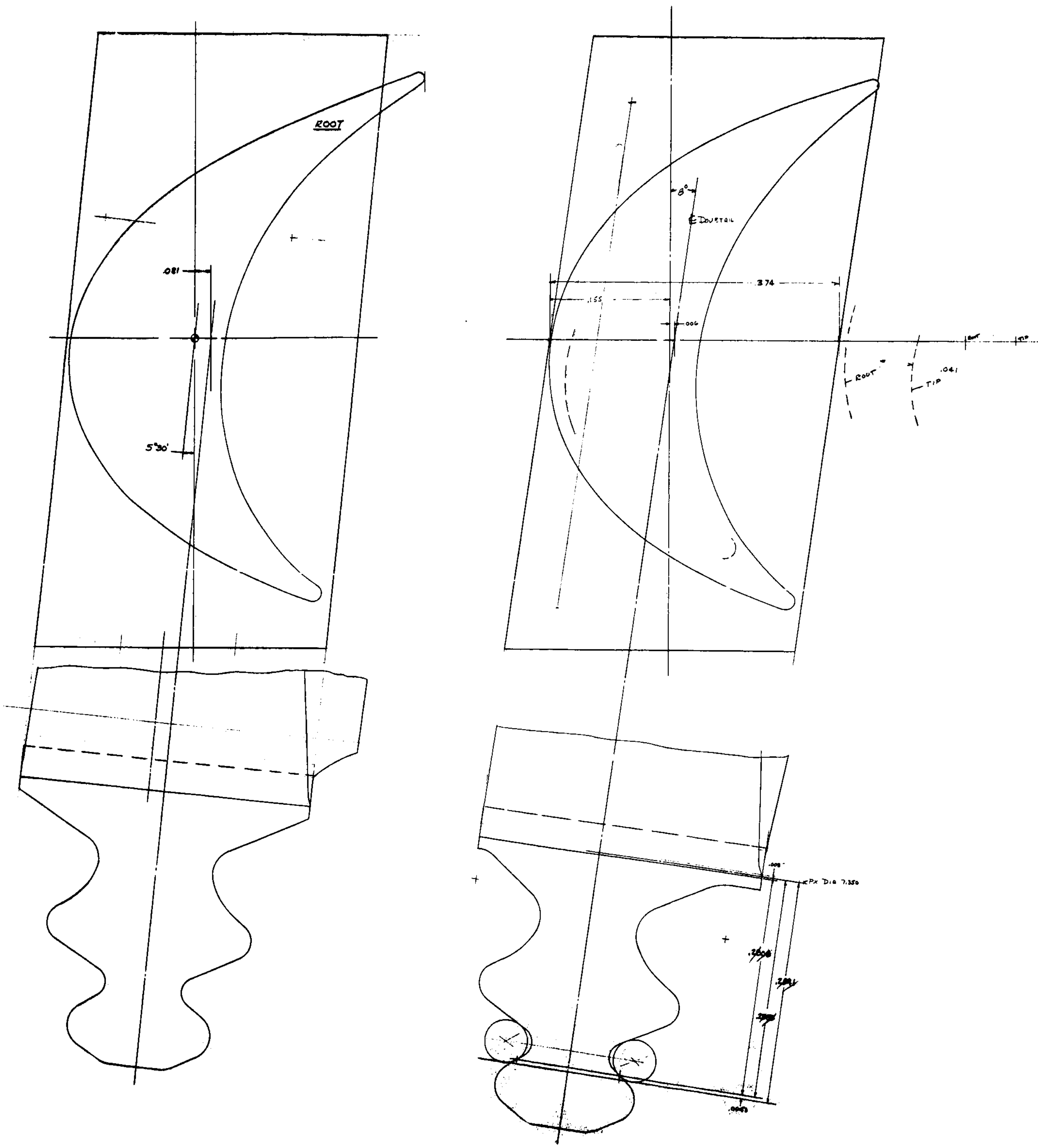


Figure 3. Comparison of Preliminary and Final First Stage Bucket Configurations

(a) Preliminary (b) Final

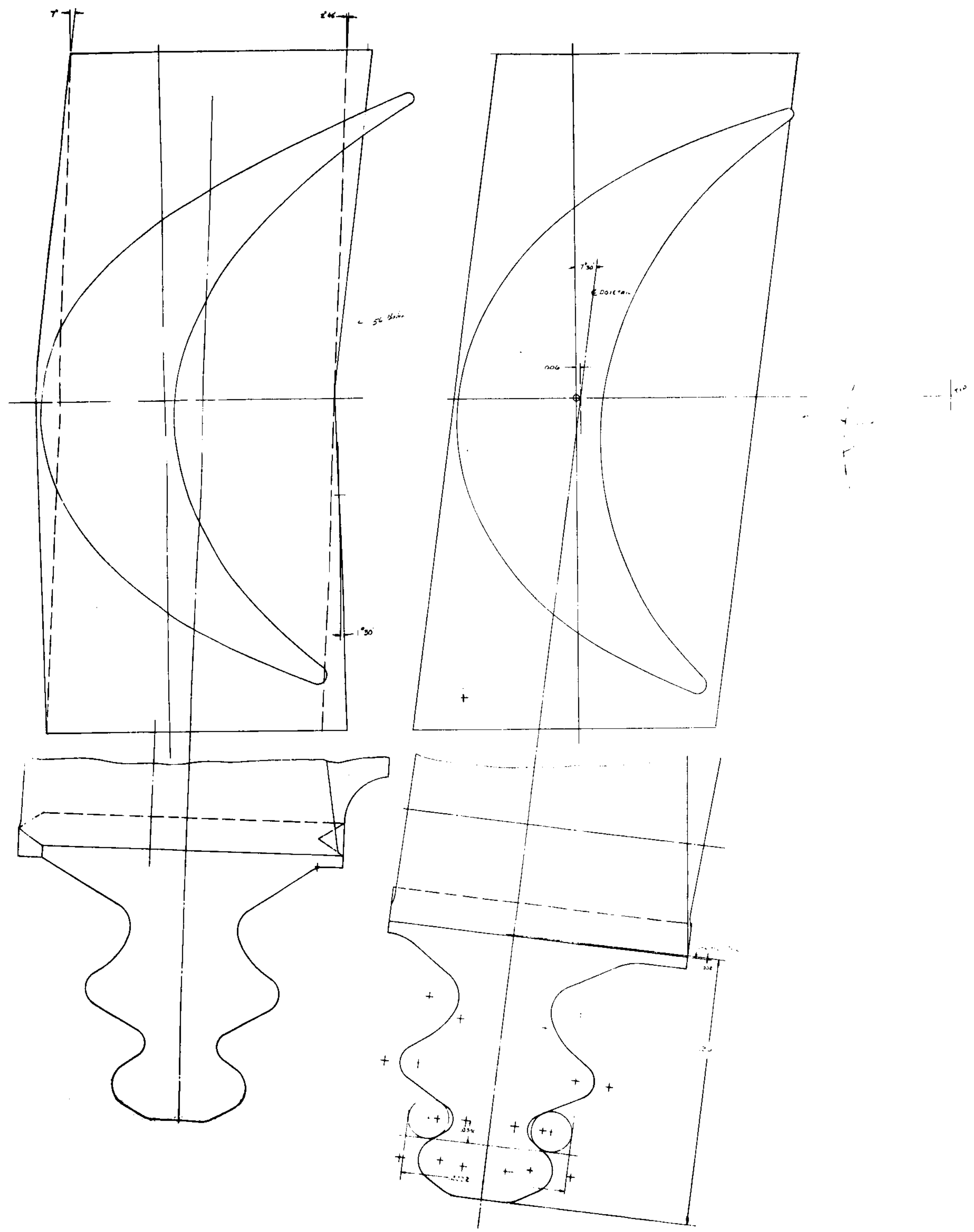
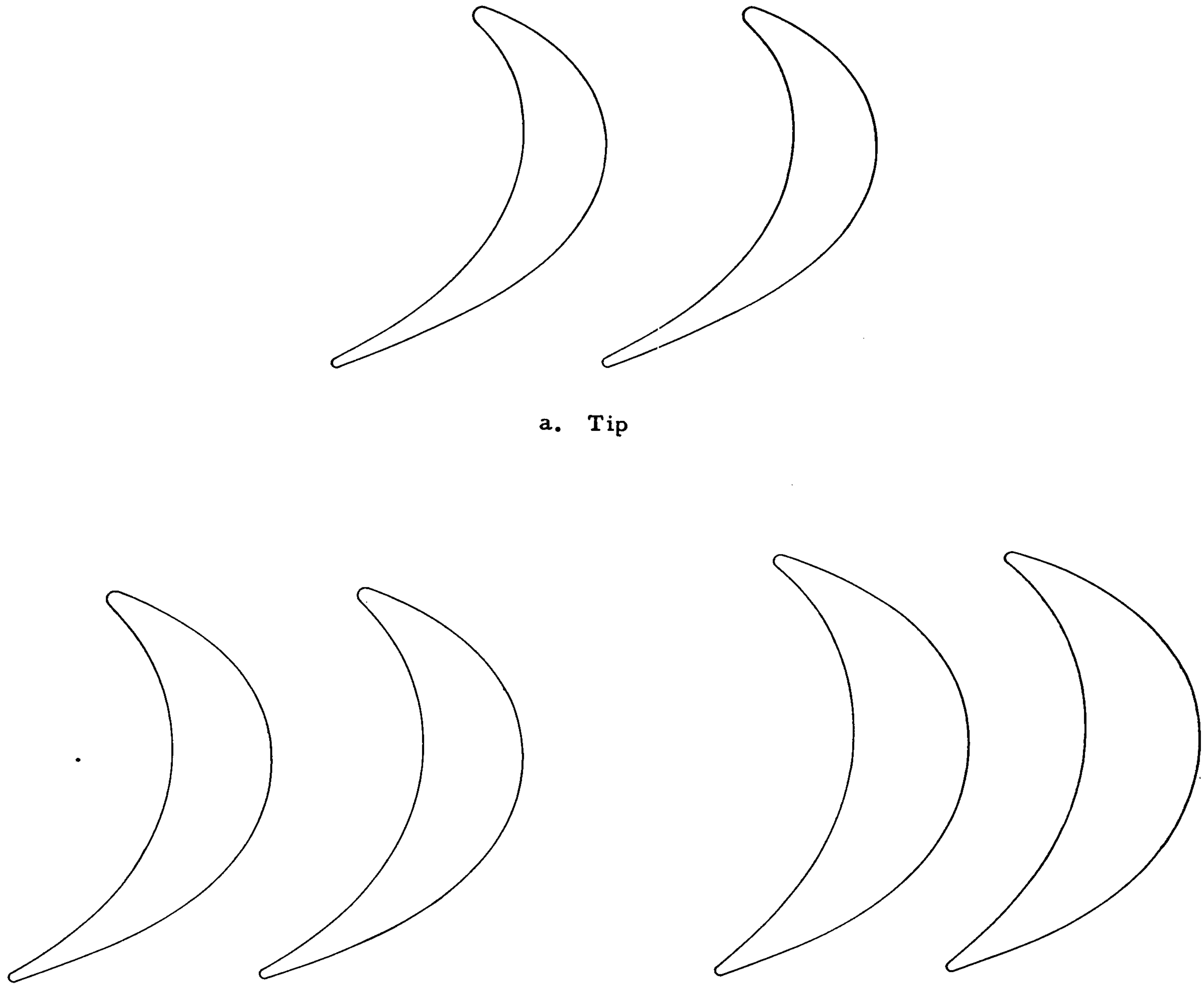


Figure 4. Comparison of Preliminary and Final Second Stage Bucket Configurations

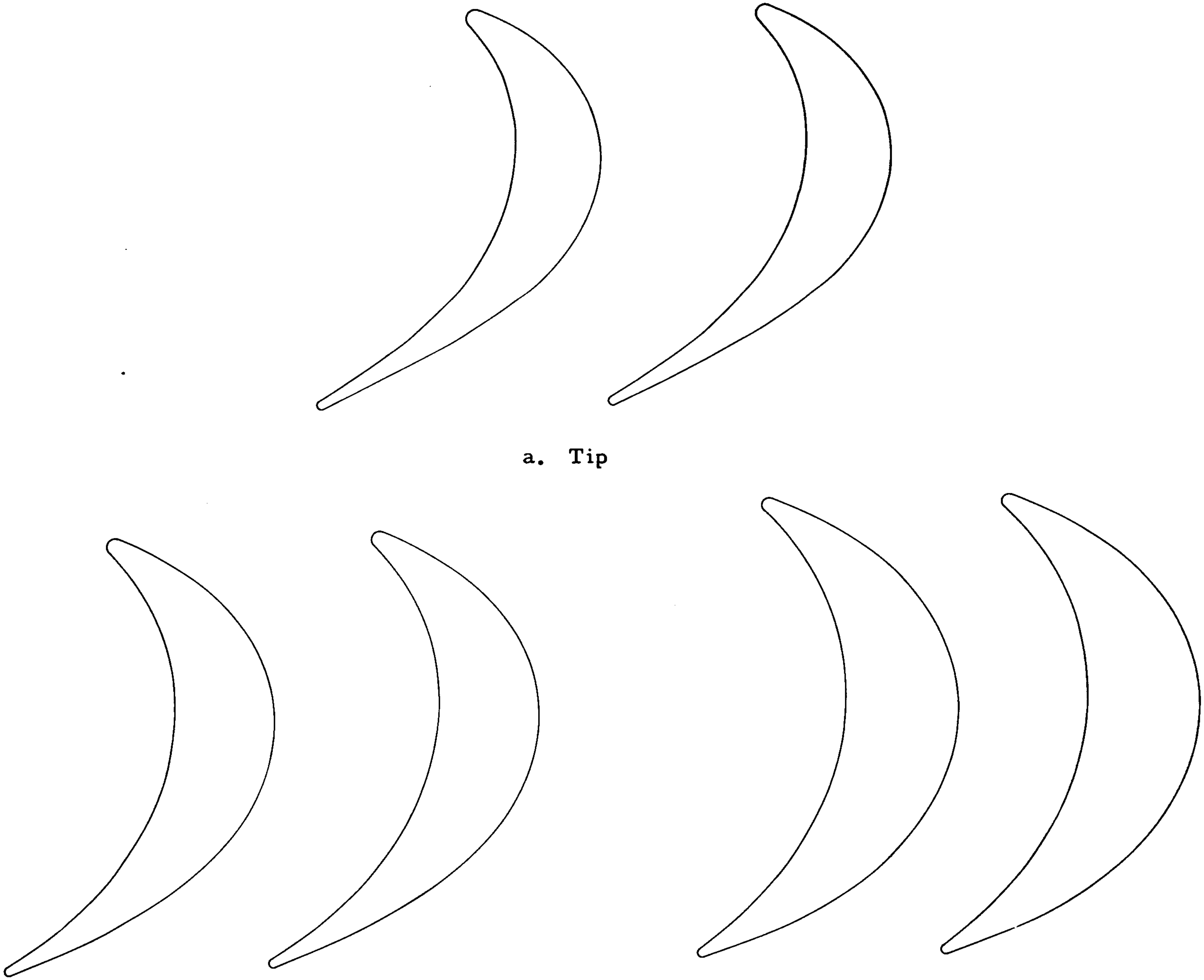


a. Tip

b. Pitch

c. Hub

Figure 5. Final First Stage Bucket Blade Sections



a. Tip

b. Pitch

c. Hub

Figure 6. Final Second Stage Bucket Blade Sections

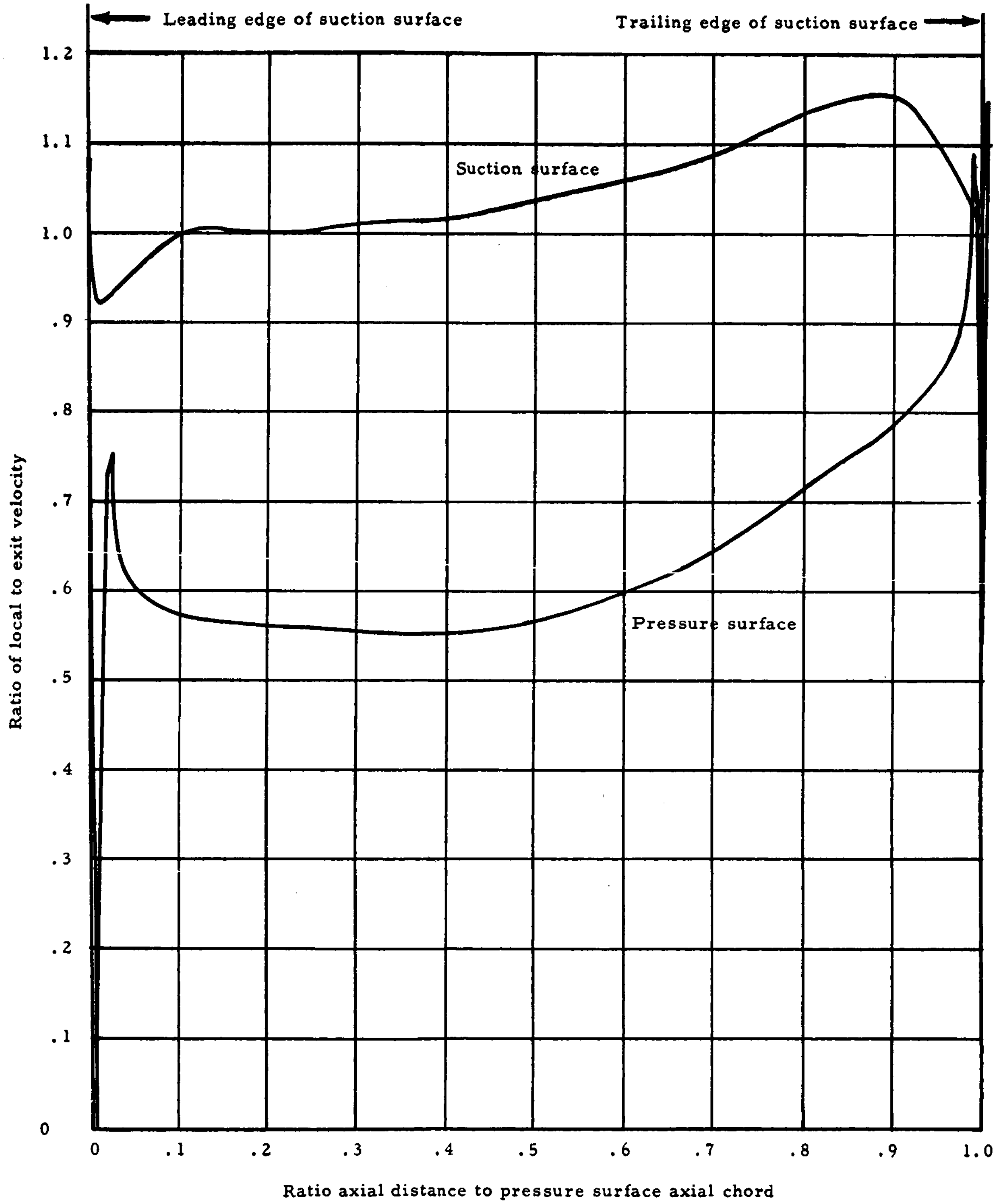


Figure 7. Velocity Distribution Around the Final Second Stage Bucket Hub Section

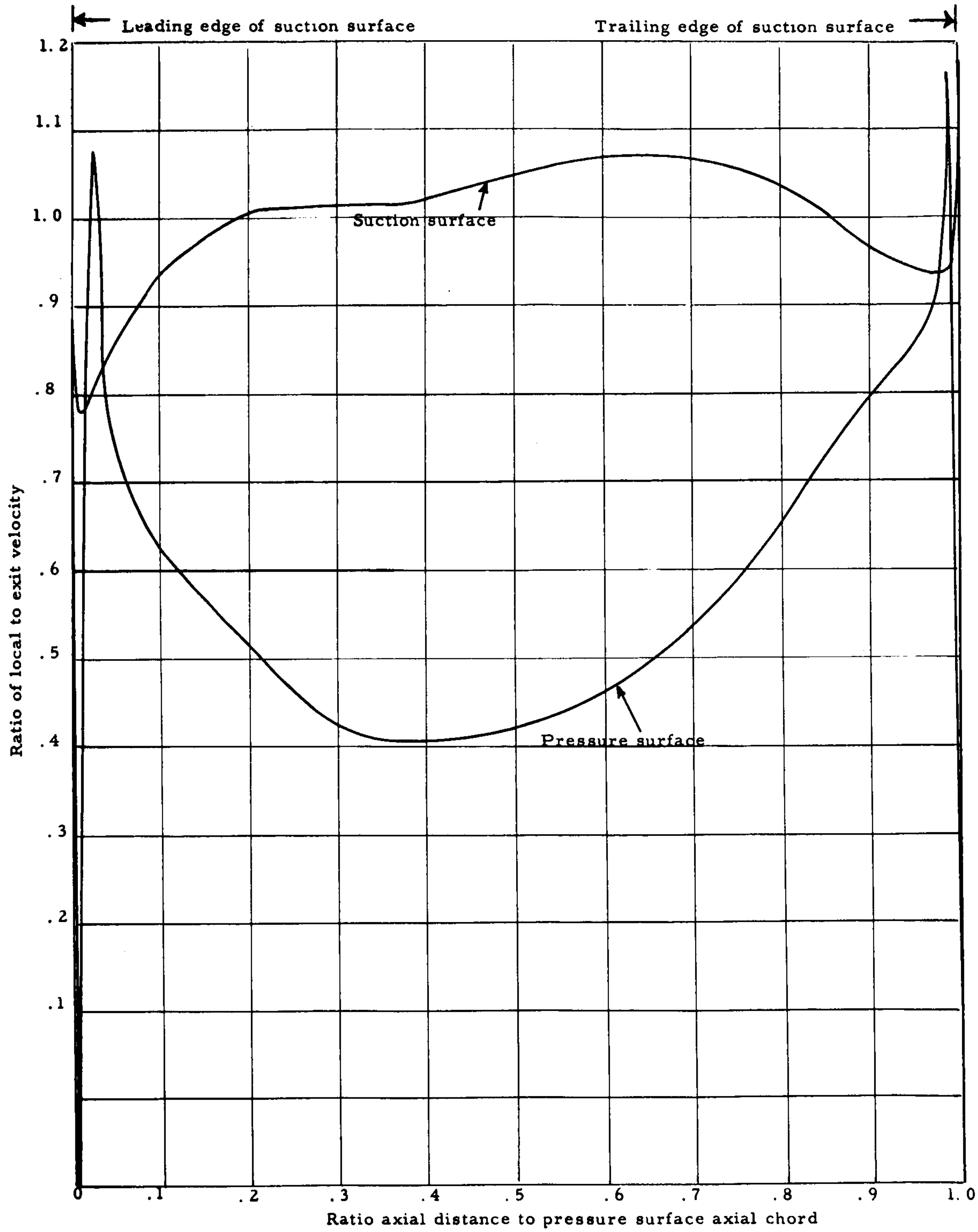


Figure 8. Velocity Distribution Around the Preliminary Second Stage Bucket Hub Section

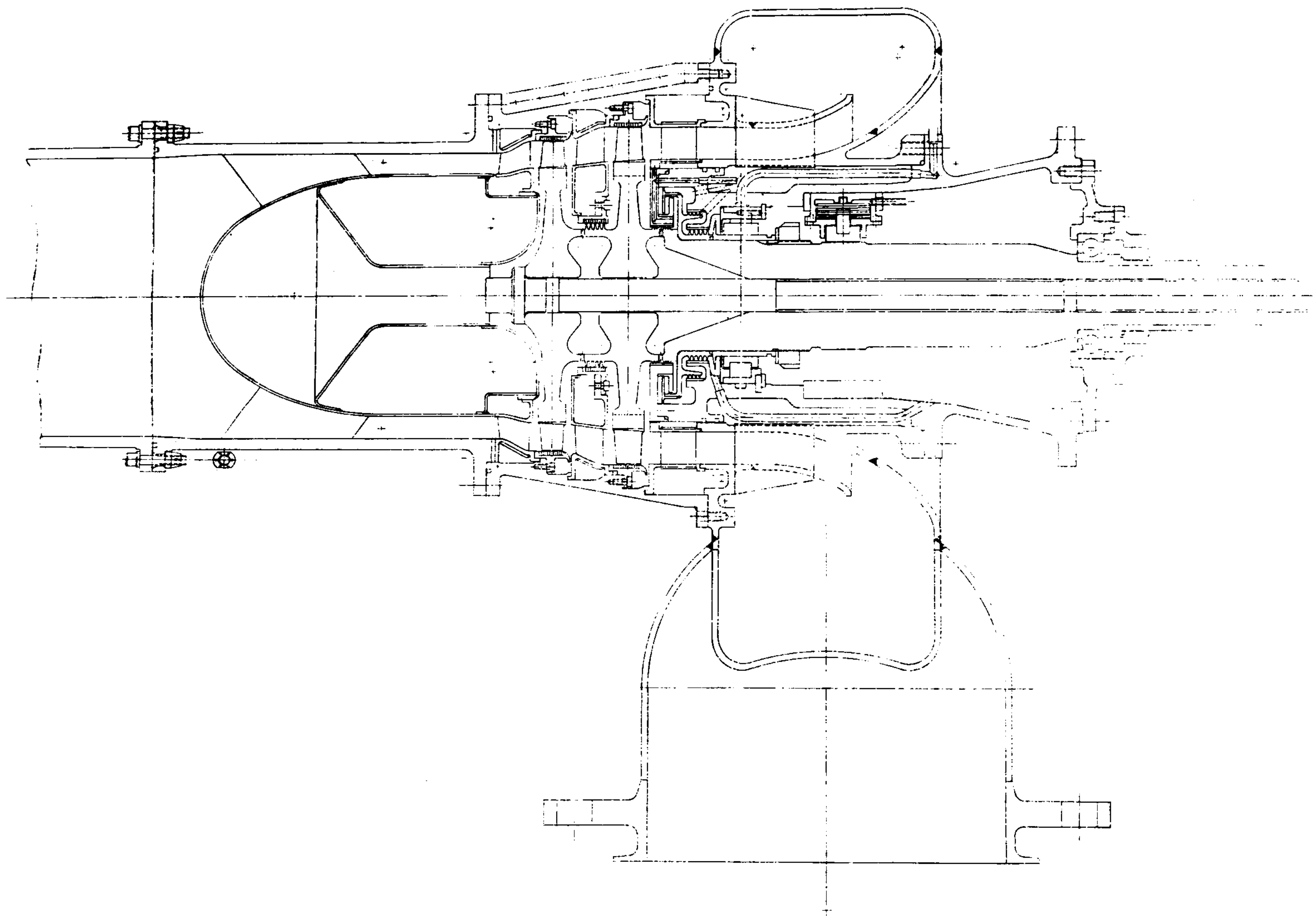


Figure 9. Test Turbine Layout Drawing

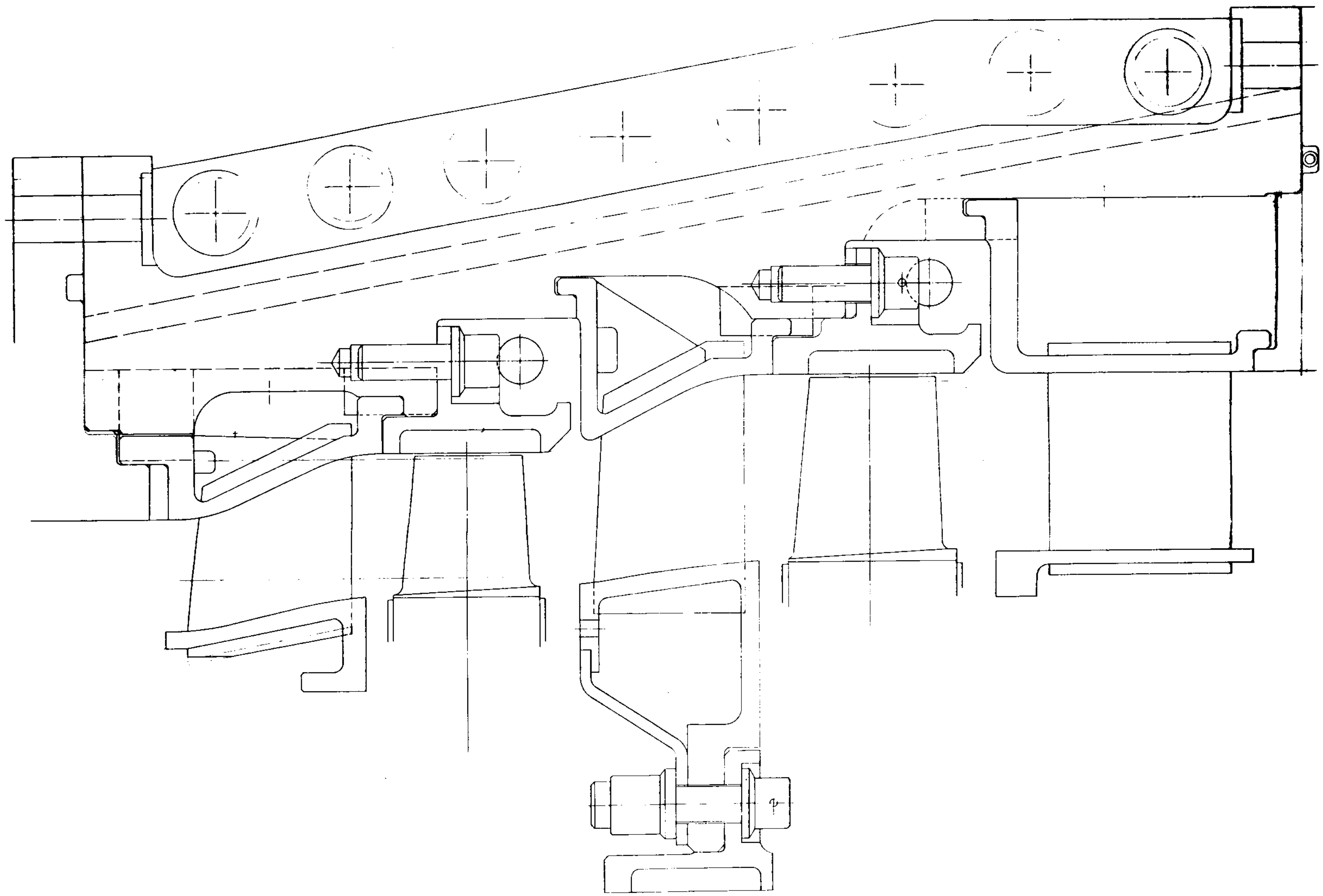


Figure 10. Rotor and Stator Cross-Sectional Drawing

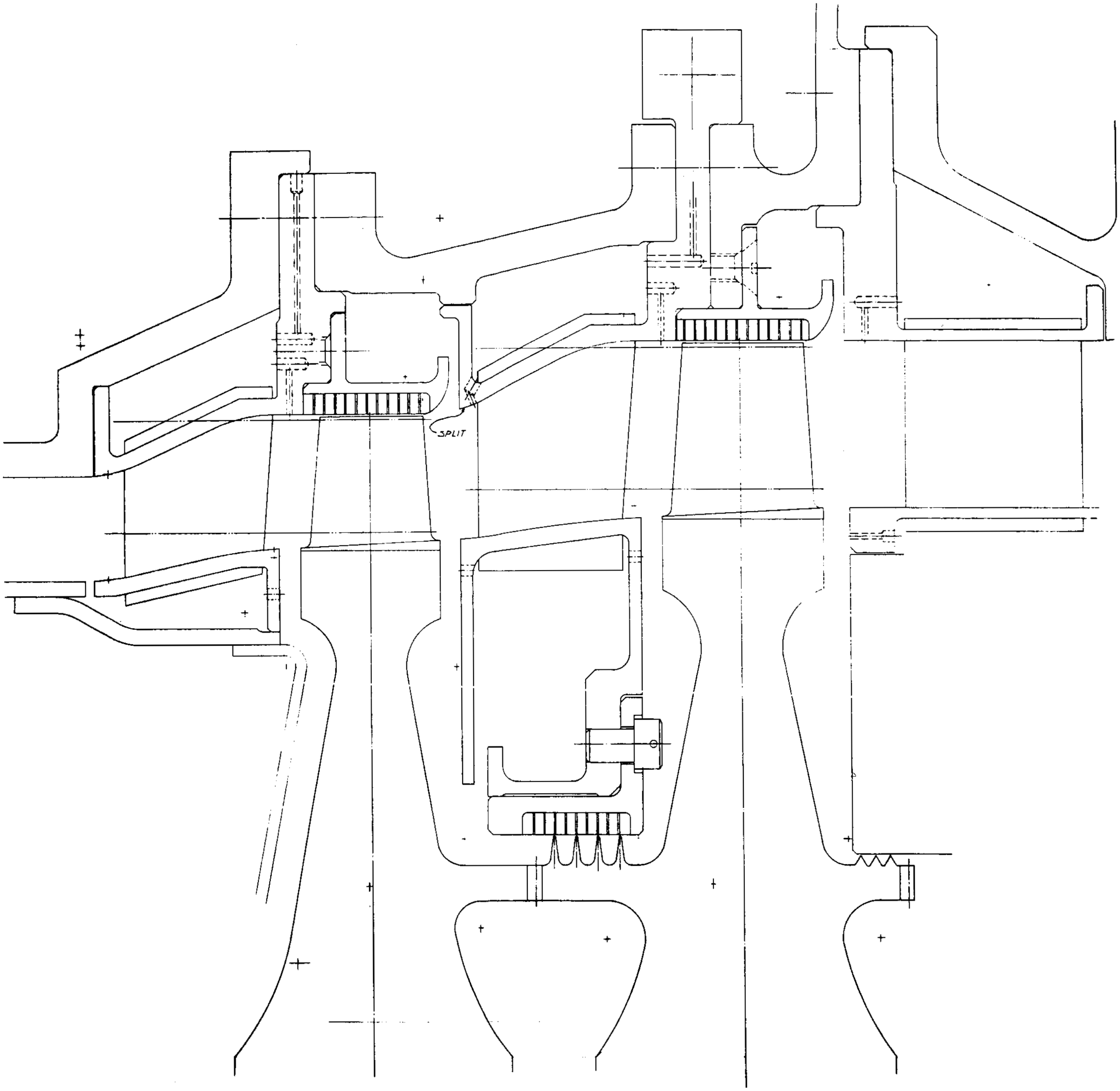


Figure 11. Drawing of Circumferentially Flanged Casing

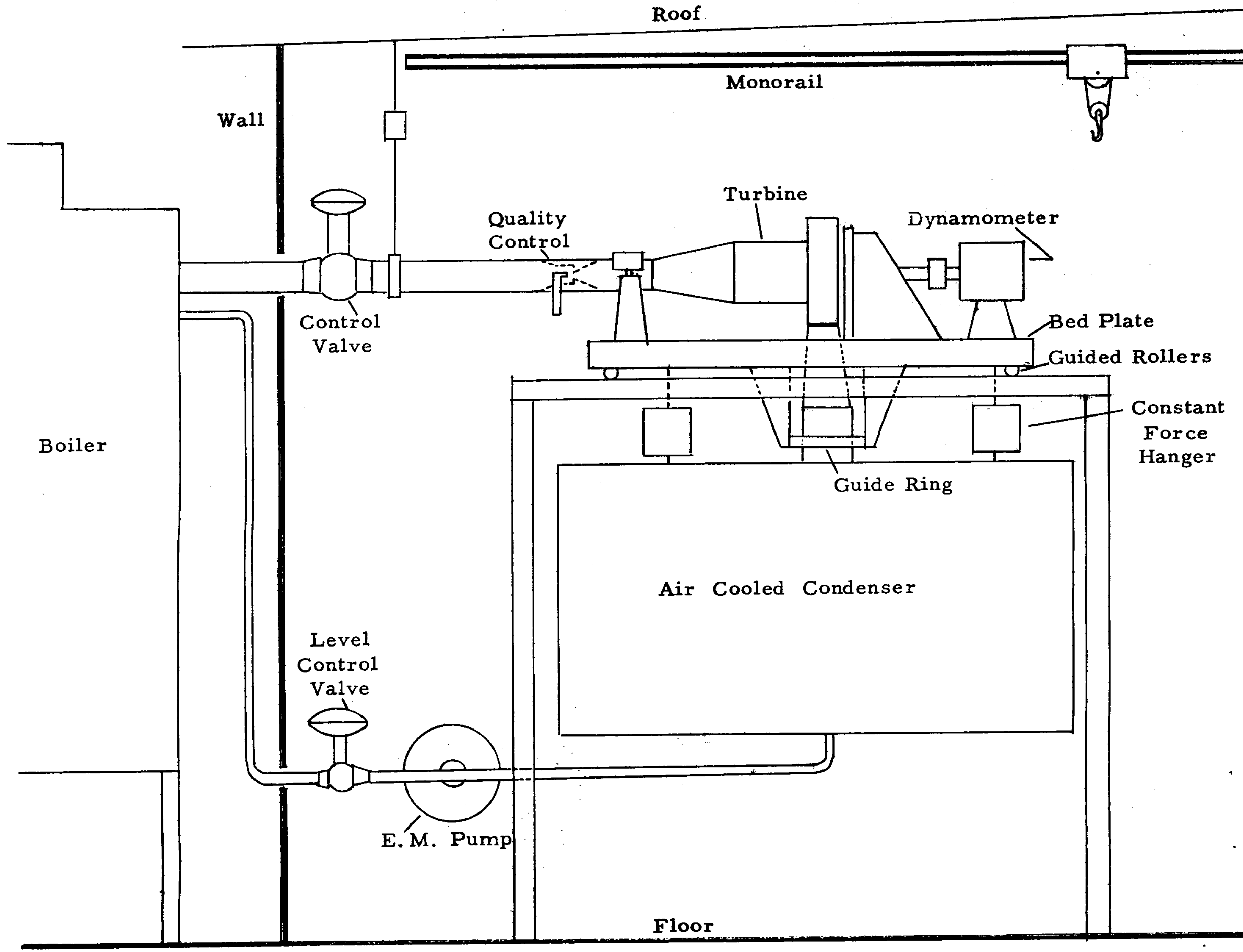
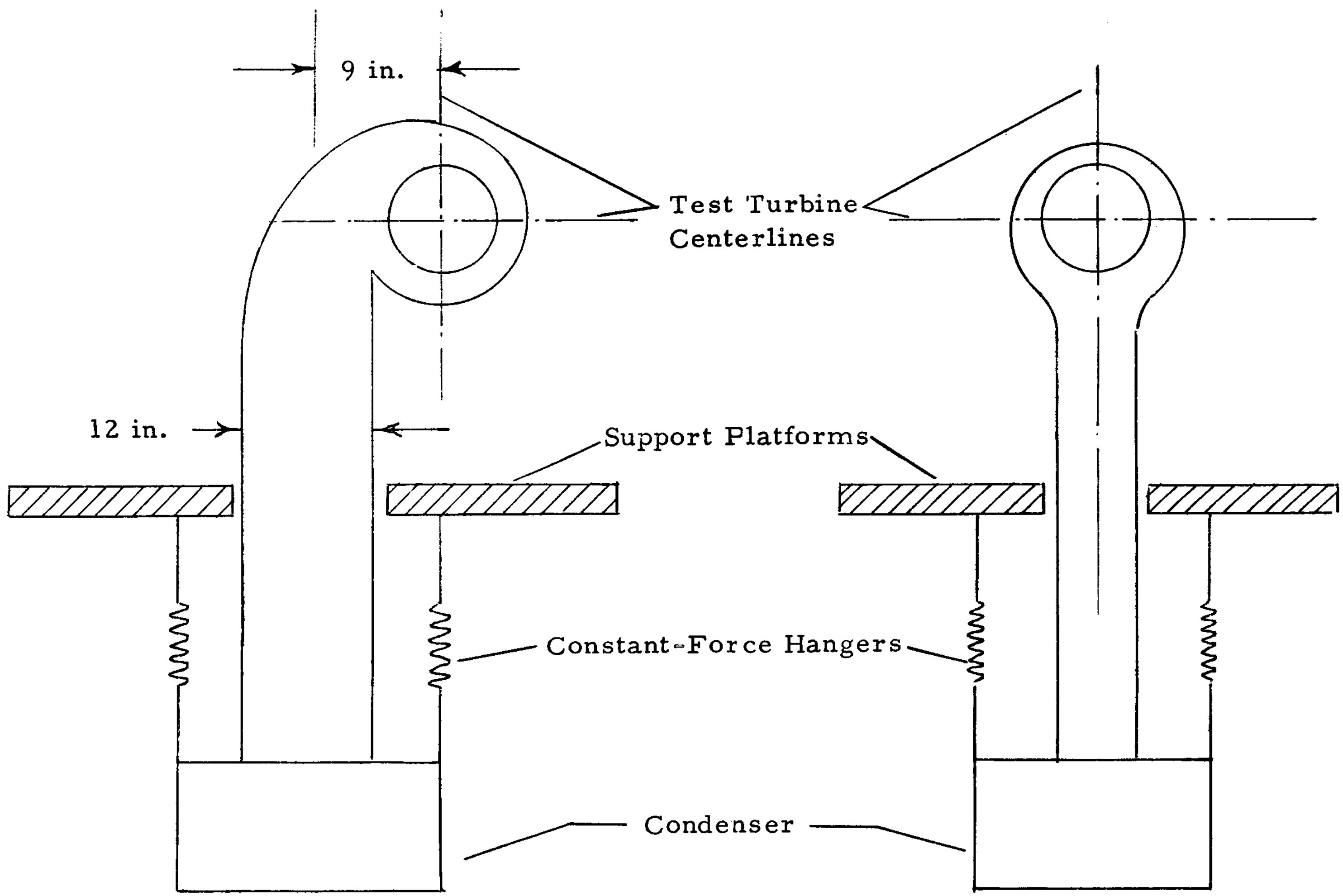


Figure 12. Schematic Diagram of Test Facility



(a) Involute Scroll (b) Symmetrical Scroll

Figure 13. Sketches of Scroll Arrangements

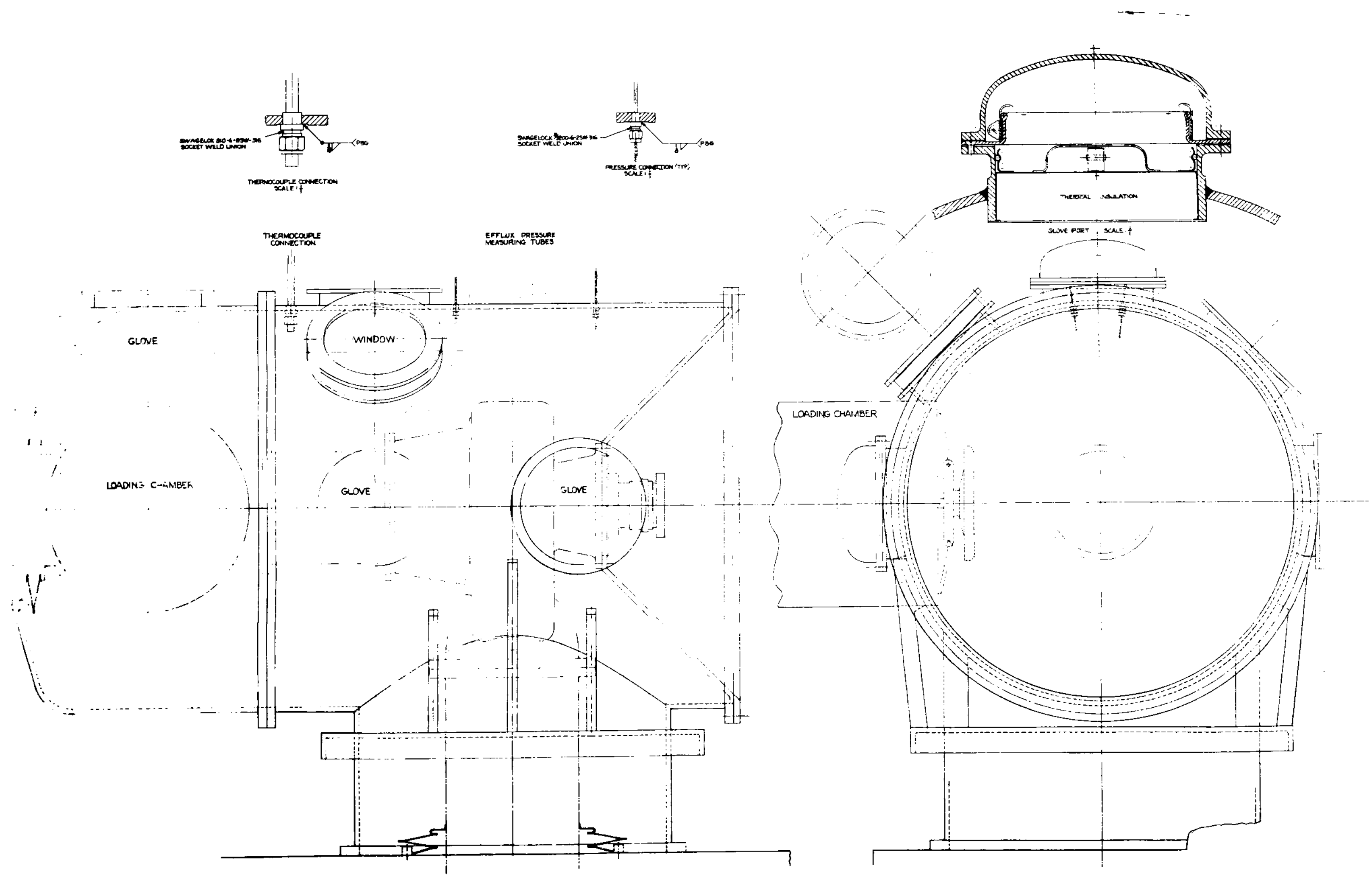
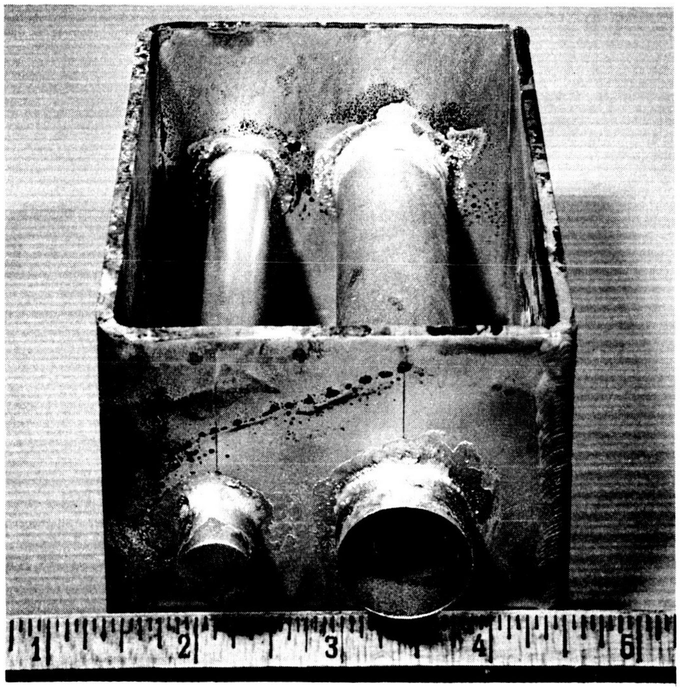
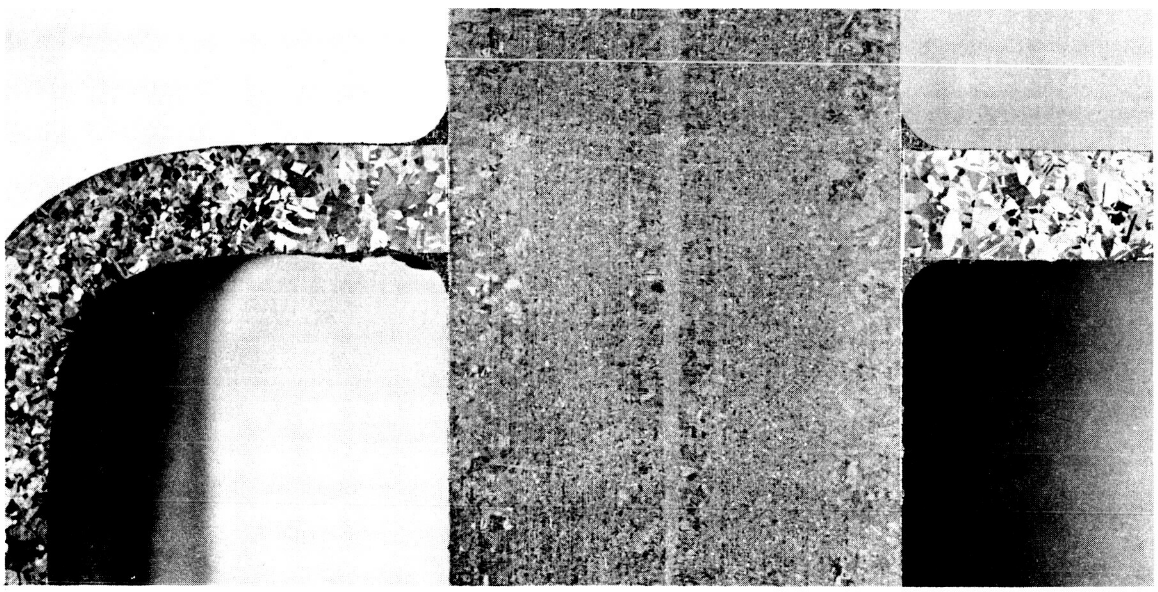


Figure 14 Glove Box Tentative Design



(a) Full View



(b) Brazed Joint Cross Sectional View

Figure 15. Nozzle Partition Brazing Test Piece

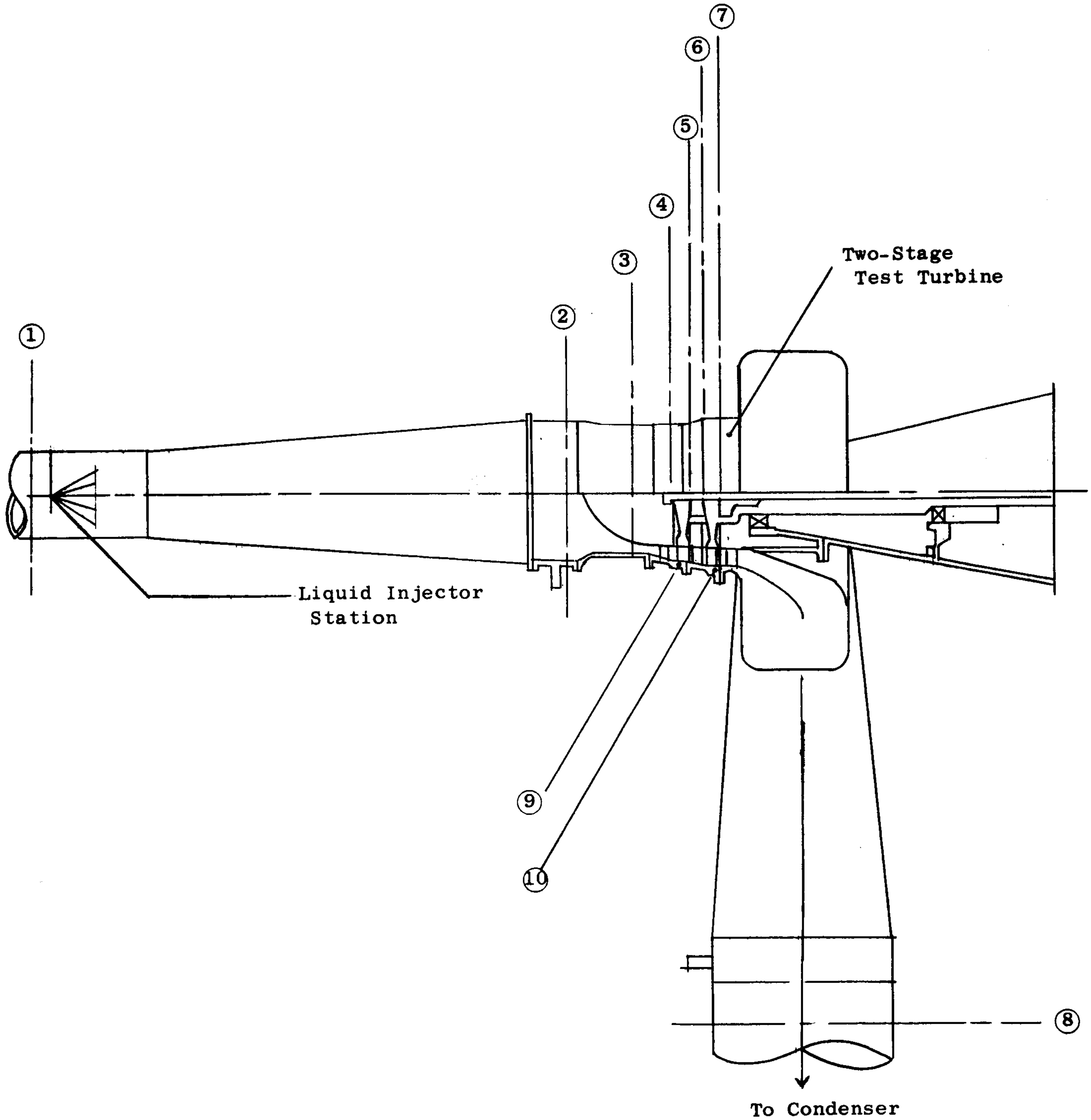


Figure 16. Instrumentation Stations

Note: All total pressure and total temperature sensors arranged in equal area annuli.

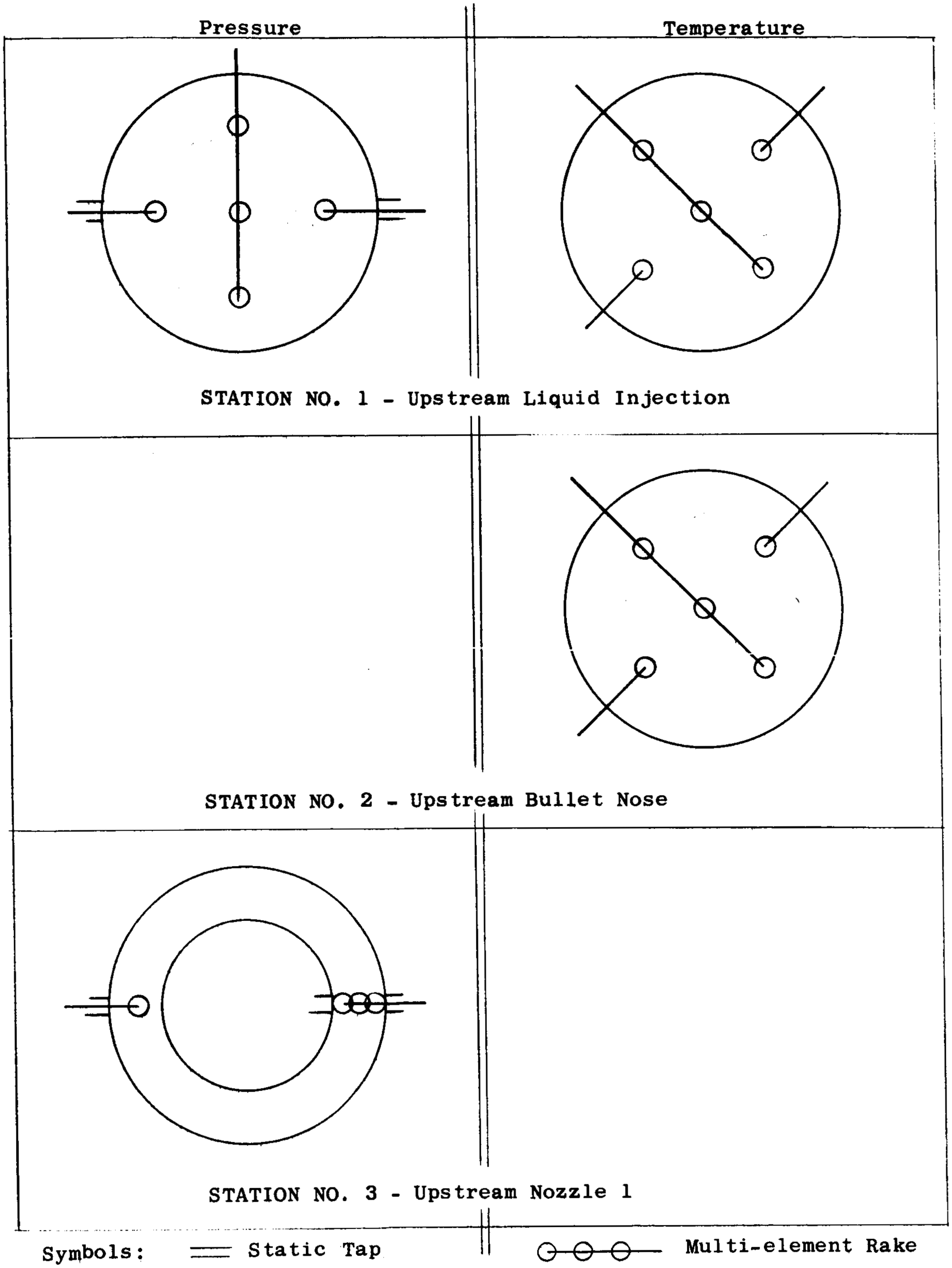


Figure 17a. Potassium Test Turbine Instrumentation Location

Note: All total pressure and total temperature sensors arranged in equal area annuli.

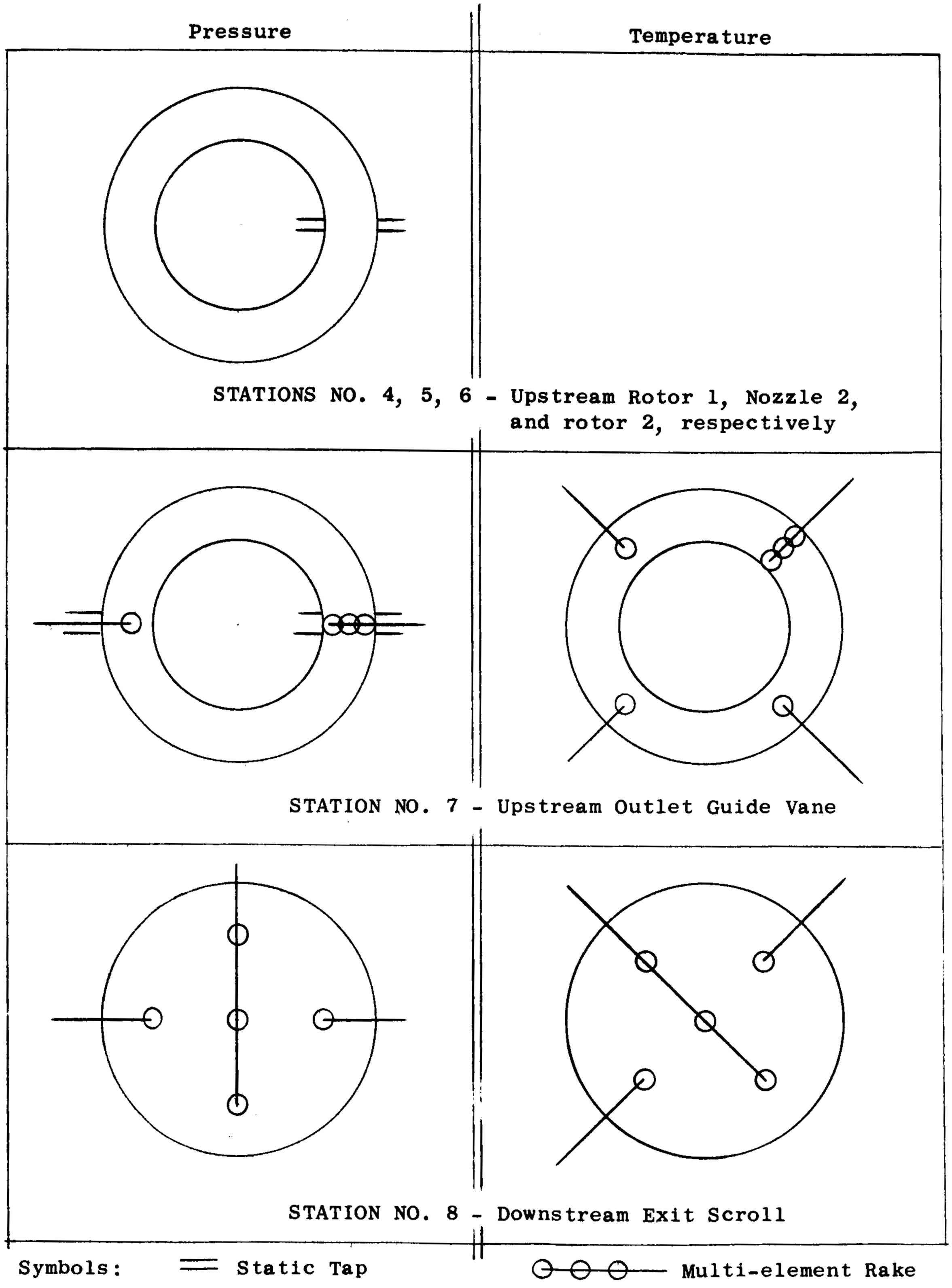


Figure 17b. Potassium Test Turbine Instrumentation Location, (continued)

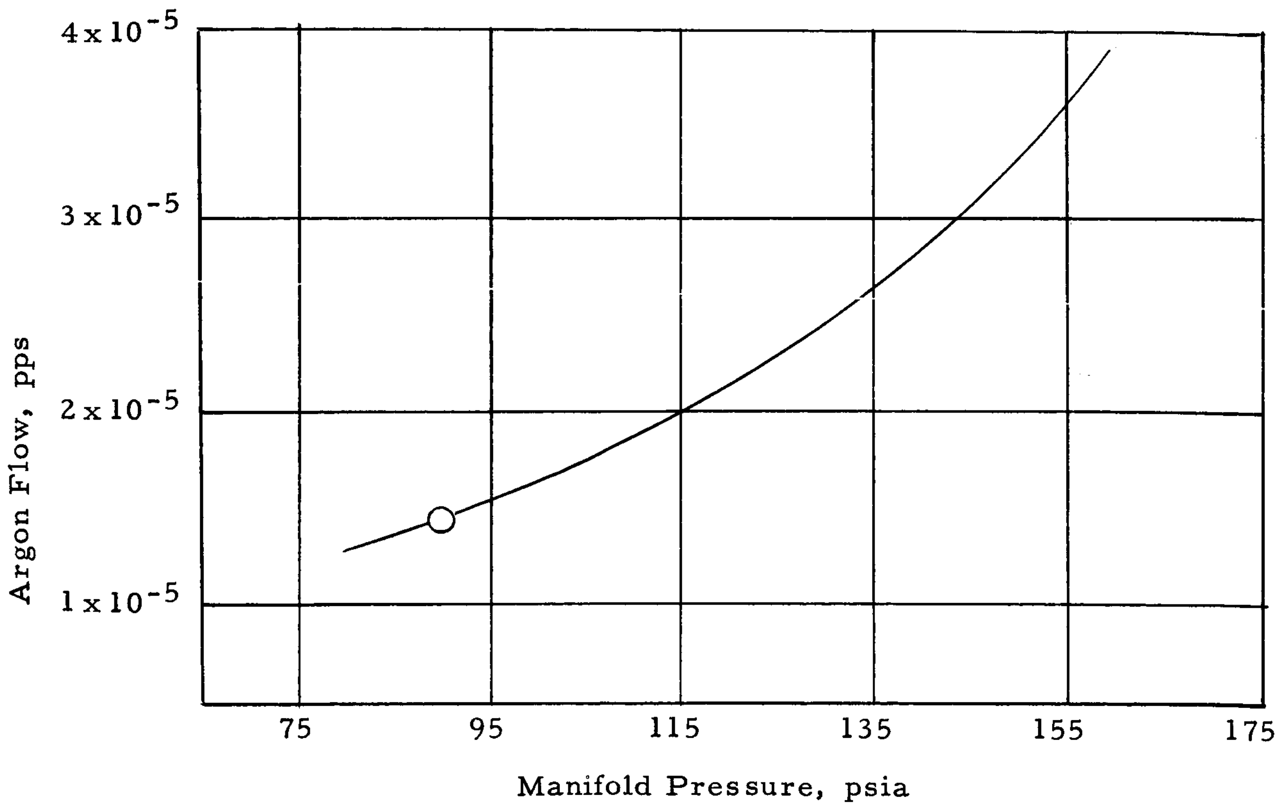


Figure 18. Experimentally Determined Normal Gas Flow From One Efflux Pressure Measuring Device. Needle Size, 0.005 in. inside diameter.

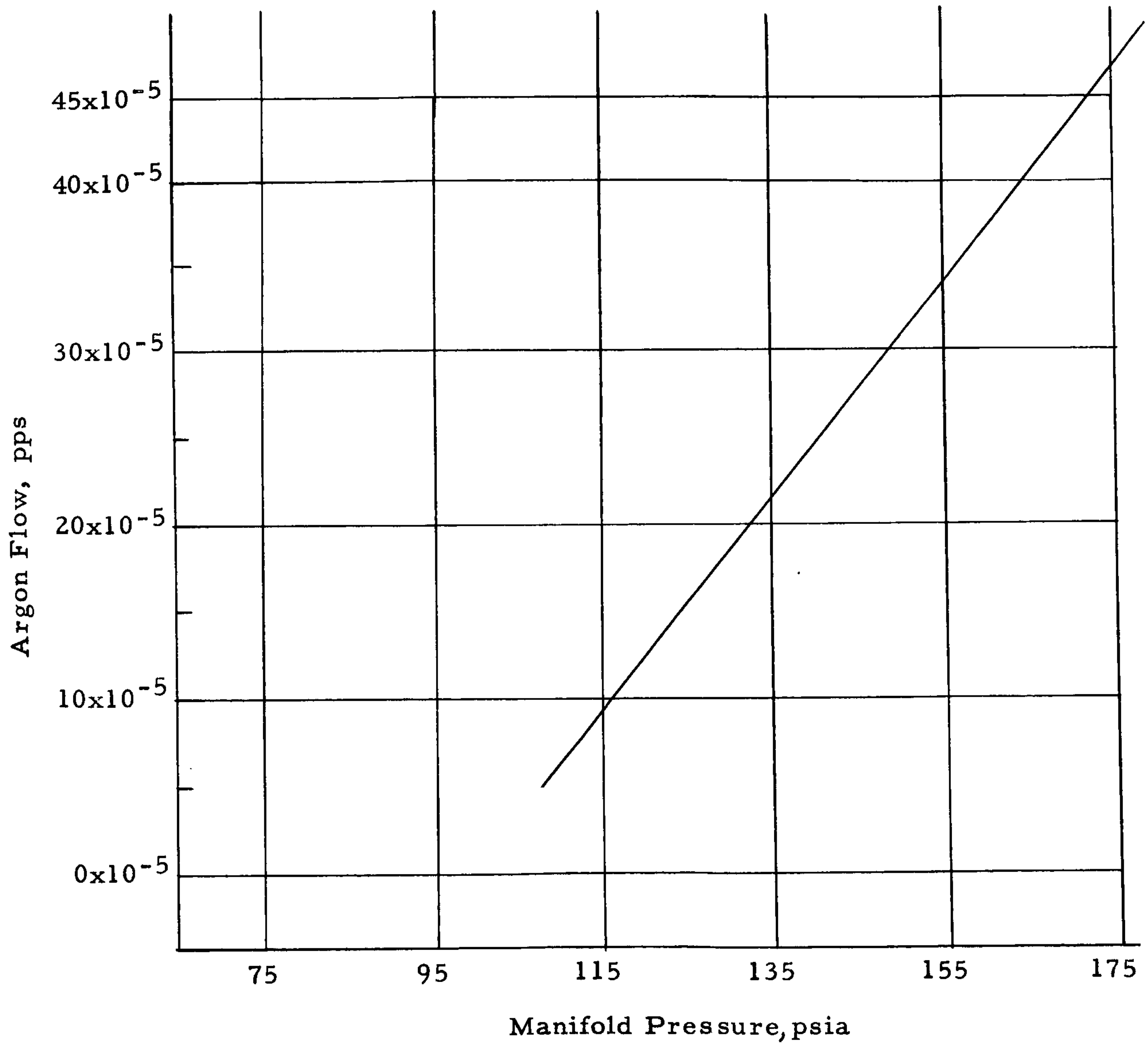


Figure 19. Experimentally Determined Purge Gas Flow From One Efflux Pressure Measuring Device. Needle Size, .016 inside diameter.

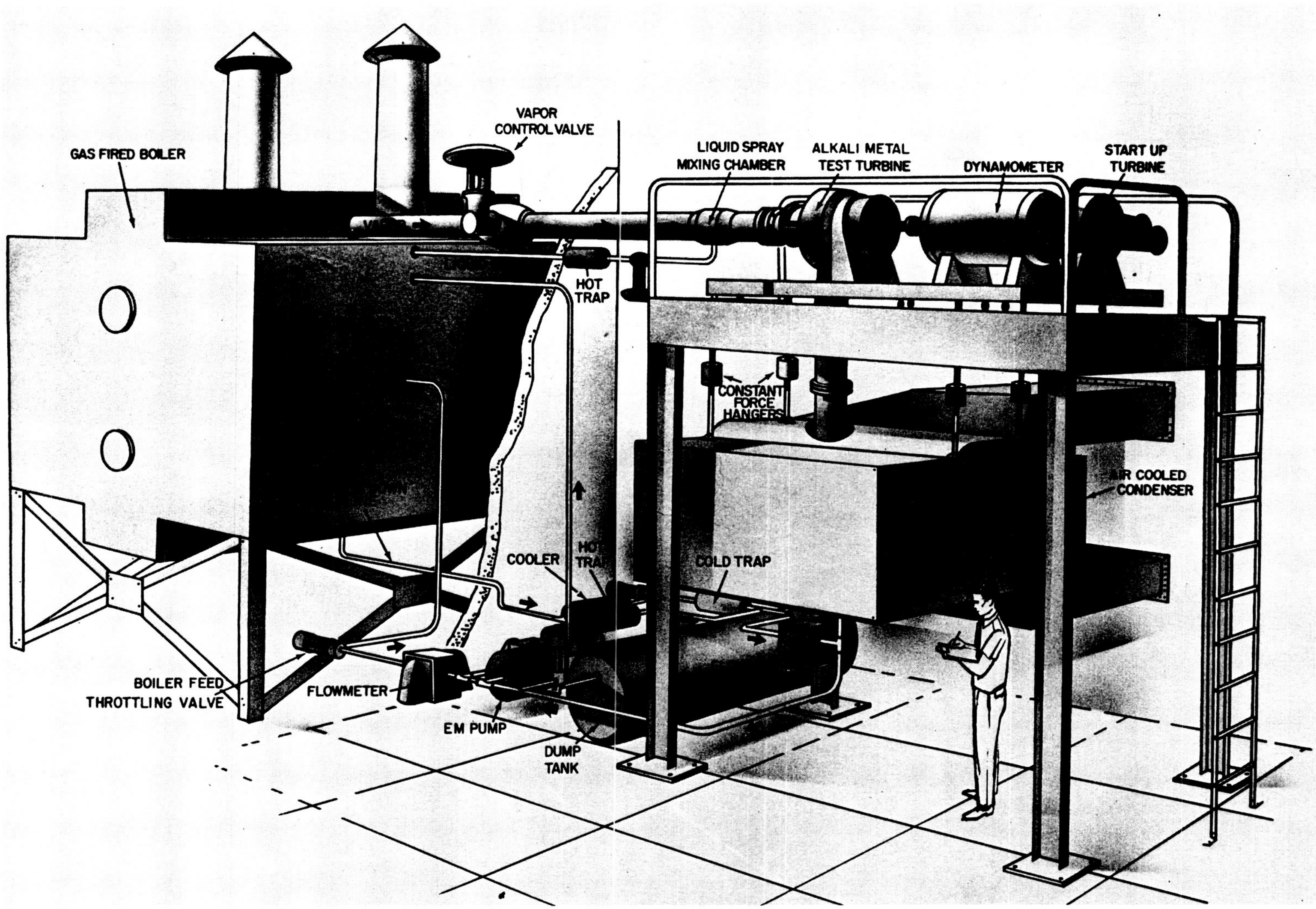


Figure 20 3000 KW Component Test Facility

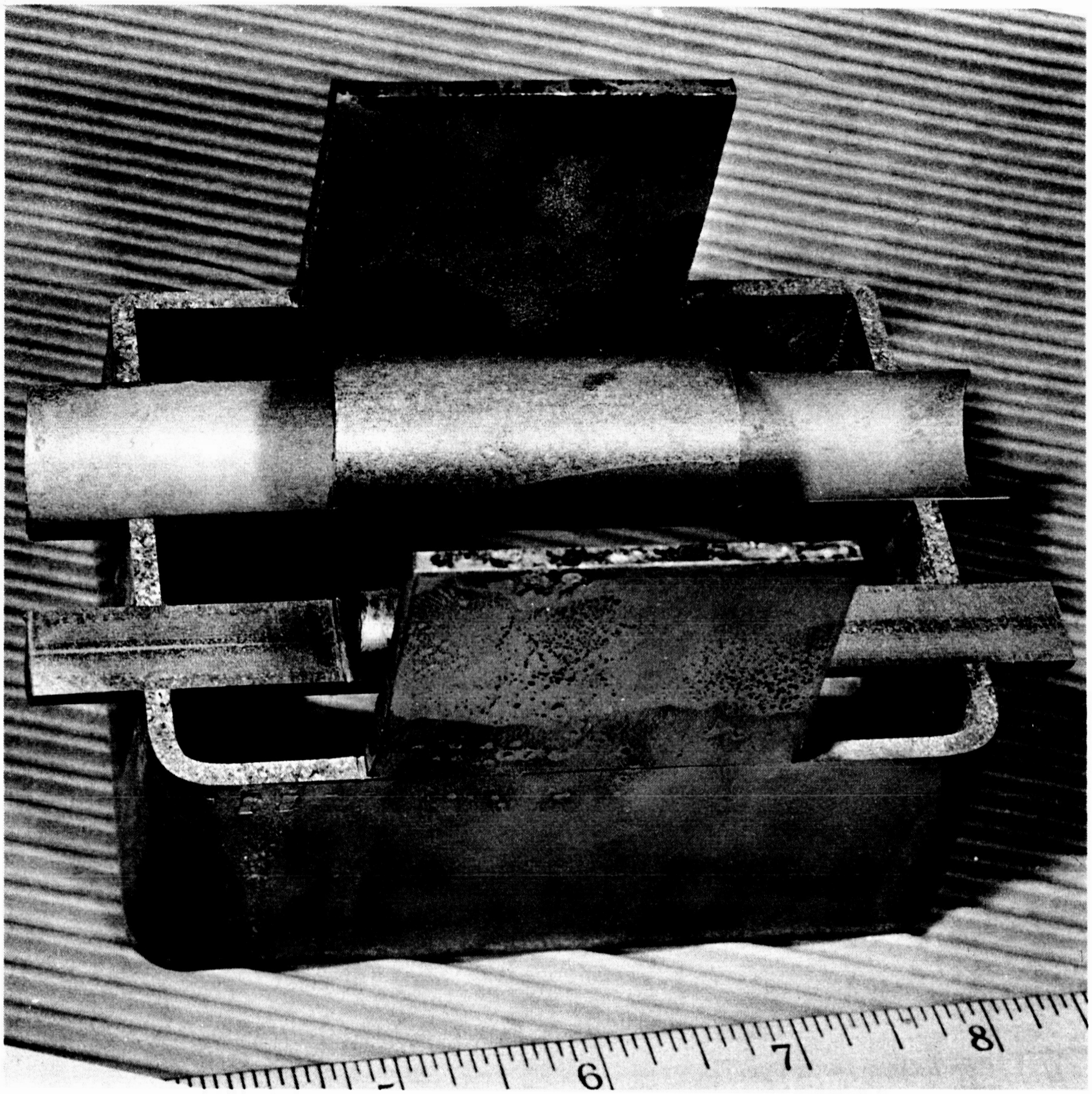


Figure 21. Sectioned Mock-Up Specimen of L-605 Bar and Tube Brazed to a Type 316 SS Frame with H-33 Brazing Alloy

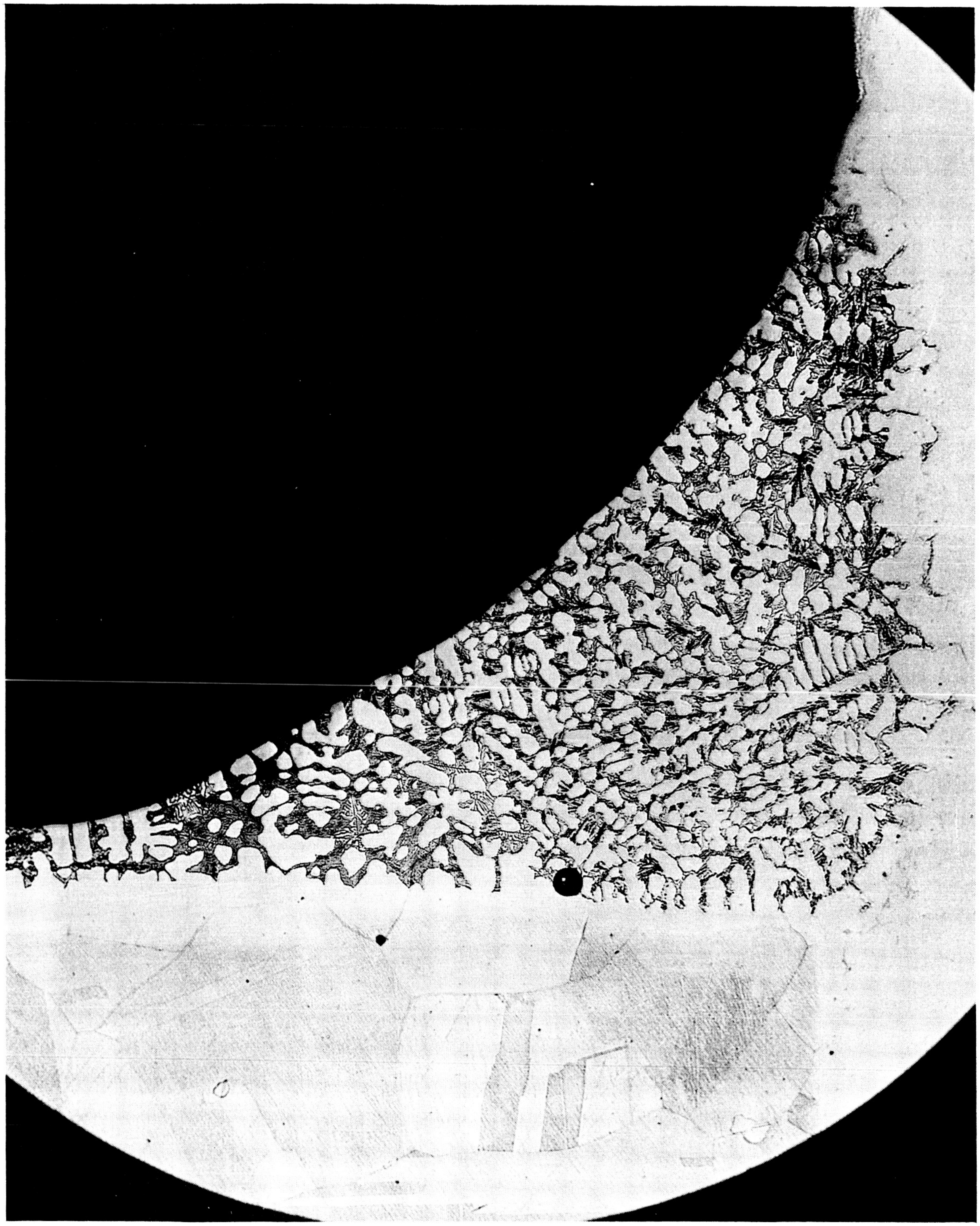


Figure 22. Cross Section of L-605 Brazed to Type 316 SS with H-33. The L-605 is at the Bottom (100X)

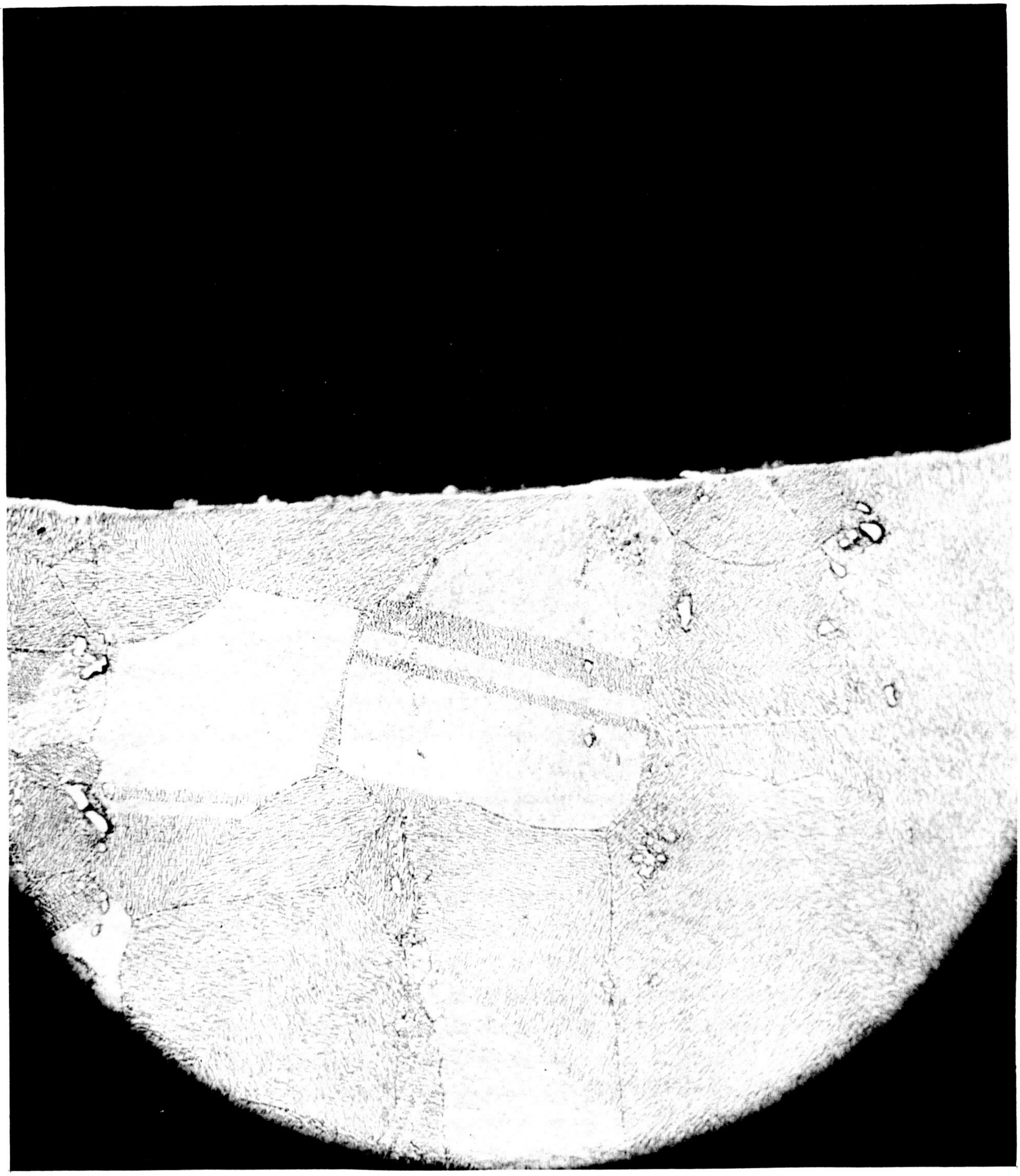


Figure 23 Cross Section of U-700 Showing the Structure at the Surface After Exposure to Potassium for 1,000 Hours at 1500°F (500X)

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